NOTES

ON

ELEMENTARY KINEMATICS

BY

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PREFACE

THE graphical course which these notes are to accompany and direct is intended to have an early place in a course in mechanical engineering, following immediately upon the work in mechanical and machine drawing. It is felt that, in the mental and graphical processes involved, this analysis of simple motions appropriately comes next after the portrayal of form. With the drafting-room work here laid out should go a light recitation course, largely descriptive, for which the writer finds Dunkerley's "Mechanism" the best text available, taking Chapters I and II, part of Chapter VII, and selected articles from other chapters.

The idea of this course is to use the most important and representative simple linkage mechanisms, have the student analyze them thoroughly as to displacement and velocity, and in connection with these examples to develop general methods and principles. The common engine mechanism, with its variations, is naturally given the prominent place. Complete diagrams, covering the whole cycle of operations of the machine, are made the aim of each case, so that the results shall be in useful and usable form. In the matter of velocity relations, the method of the instantaneous center is given decided preference, as the best foundation upon which the student shall build his concept of these relations.

In general, the idea has been to lay out a simple, practical course, not covering a very wide range of mechanism, but aiming to set forth the best and clearest methods of work. The kinematics of more complex linkages and of the higher-pair mechanisms—cams, non-circular wheels, and the like—is entirely deferred, except for a brief study of the standard gear-tooth

profiles. The brevity of the treatment of this subject, in section K, is due to the fact that this is merely intended to supplement another text, and to give certain definite directions for graphical work. In the Appendix will be found the detailed layout of the course, and a few supplementary problems.

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NOTES ON ELEMENTARY KINEMATICS

A. Introduction

1. The Kinematics of Machinery is the science of constrained motion. Constraint of motion is an essential characteristic of a machine; by it we mean that the several parts or members are compelled to move in a certain definite, unchangeable manner relative to each other. Whatever forces are needed to hold any moving piece in its predetermined path are supplied by the other parts with which it has contact; and with proper design each part must have ample strength and rigidity to perform its share in maintaining this definiteness of motion.

2. A machine and its parts must meet two requirements; they must have definite form and sufficient strength to maintain that form under the action of force. In a kinematic discussion we assume the existence of needed strength, and then pay no further attention to that side of the matter. With emphasis thus laid on form and motion, the term "mechanism" is to be used rather than "machine." A mechanism is the form-skeleton of a machine; a machine is a mechanism embodied and put to work. In practical usage, a machine is called a mechanism when the motion side of its activity is the more prominent or the more important.

3. The term "motion," used in the most general sense, covers three manifestations, or phases, or degrees. The first is change of position, or displacement; the second, rate of change, or velocity; the third, change in velocity, or acceleration. With freely moving bodies—most perfectly instanced in celestial

mechanics—we must begin with force, and reason through acceleration and velocity to path. With constrained motion, the line of reasoning is reversed. The same fundamental relations between force, mass, and motion hold in either case; but in the second we may begin with a prescribed path, and can solve a great many problems of motion by simple geometrical methods.

4. There are two kinds of constraint in machinery. The first is form constraint, which has just been defined, and which fully determines the relative motion of machine parts. The second, which is exerted in many working machines, is velocity constraint. The most obvious example is the action of the flywheel in the ordinary reciprocating engine, which compels the crank to rotate at a practically constant rate. When this condition is imposed, not merely relative but also absolute motion becomes determinable. The sense in which the word "absolute" is here used will appear as the subject is developed.

5. It must be understood that there is no such thing as perfect constraint. No solid body is absolutely rigid and no moving joint can be altogether tight, nor will any mass move at a perfectly constant rate under the action of a pulsing force. But in disregarding elastic yield and vibration or lost motion in joints or the minor fluctuations in the speed of an engine shaft, we simply assume that in the problem under consideration they are of negligeable magnitude. In other discussions, such actions may be of the first importance.

6. The problems in the following course are concerned with displacements or movements, and with some of the simpler velocity relations. Simple mechanisms of wide application are selected for analysis, and in the solution of these concrete problems general principles and methods will be developed. The first mechanism taken up is that of the common reciprocating engine, in which the fundamental problem is a change from straight-line to rotary motion.

B. The Crossed Slider-Crank; Harmonic Motion

7. In Fig. 1, let OC be a radius rotating about the center O, so that the point C travels in a circular path. The simplest straight-line reciprocating motion which can be derived from the motion of C is that of the point D along the diameter AB—D being the continual projection of C upon AB.

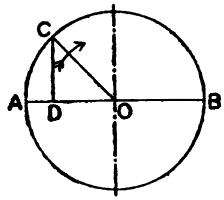

FIG. 1.—Fundamental Diagram.

In Fig. 2 a mechanism is built around the geometrical diagram of Fig. 1. The parts or members or links of this mechanism are as follows:

The fixed frame or body, piece 1;

The rotating crank OC or 2, which turns in frame 1 at O and carries a crank-pin at C;

The slide-block 3;

The crossed slide 4; the whole slide SS works in guides on the frame 1, while in the cross-slot TT the block 3 slides up

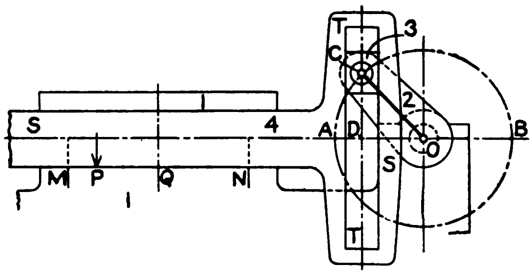

FIG. 2.—The Crossed Slider-crank.

and down. Slot TT is perpendicular to slide SS, hence the whole piece 4 has the same motion as the projected point D in Fig. 1.

To show this movement of the slide 4, as distinct from that of the crank which produces and determines it, we may mark a vertical line at P on the slide-bar, and refer it to fixed marks at M and N on the guide. The three lengths DP, AM, and BN

(measured parallel to the stroke-line) are made equal: then MN is the stroke or stroke-length of P, and this stroke is equal to the diameter of the crank-circle. It is evident that D travels and is located on AB just as P on MN: so far as this mechanism by itself is concerned, D is the most convenient point on piece 4 to be used in measuring the movement of that piece; the reason for locating P on MN lies in the analogy to what must be done with the actual engine mechanism.

The device shown in Fig. 2 is called the crossed slider-crank mechanism. If we impose the additional requirement that the crank OC shall rotate at uniform speed, the slide 4 will have what is known as harmonic motion.

8. In Fig. 3 is given a slide-position diagram for Fig. 2. Of course, the projection of C, at D, upon AB determines the position of the slide upon its stroke-line: but by drawing the limit lines A_1A_2 and B_1B_2, perpendicular to AB and tangent to the crank-circle, and then passing through C the line EF, we have C located on EF just as D is on AB. This construction is no more convenient than the projection of C to D; but again, analogy to the diagram for the actual engine, Fig. 6, is the influential consideration.

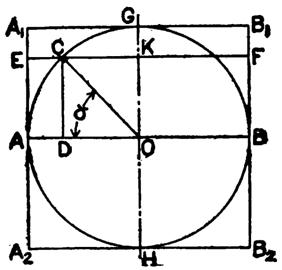

FIG. 3.—Diagram of Slide Positions.

Let the length of the crank CO be R, and let α stand for the crank-angle AOC, taken from the horizontal AO as origin: then if the slide-position be measured by its distance from the middle of the stroke, the value is

$$DO = t = R \cos \alpha; \quad . \quad . \quad (1)$$

while if, as is more usual in considering the movement of an engine piston, the travel be measured from the beginning of the stroke, at A, the value is

$$AD = s = R(1 - \cos \alpha) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This mechanism has comparatively few direct applications, but is of importance as the simplest case of the derivation of

reciprocating from rotary motion; in many approximate discussions, its simpler relations are used instead of the more complex exact relations for the actual mechanism.

C. The Engine Mechanism

9. The usual form of the mechanism of the piston engine is drawn in skeleton outline in Fig. 4. The links are, frame 1, crank 2, connecting-rod 3, and slide 4. Shown here as a plain block, piece 4 stands for the whole combination of crosshead, piston-rod, and piston in the actual machine. The three turning joints in this mechanism are, shaft bearing at O, crank-pin at C,

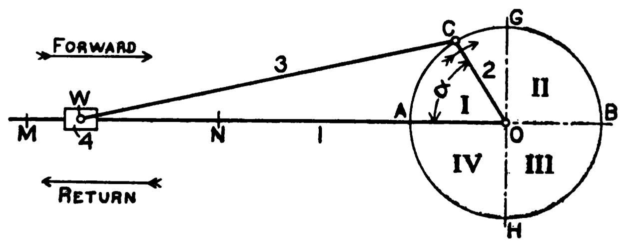

FIG. 4.—Skeleton of the Engine Mechanism.

and wrist-pin at W; and there is one sliding joint. This is called the slider-crank mechanism.

Crank travel, or the crank-angle α, is estimated from OA all the way around the circle, the latter being divided into quadrants as shown. Right-hand or clockwise rotation will be taken as the standard direction of turning, and the piston movement is divided into forward stroke, from M to N, and return stroke, from N to M. When the crank is at A and the slide at the left-hand end of its stroke (the piston being near the plain cylinder-head), the engine is said to be on its head-end dead center; the other end, with the crank at OB, is the crank end. A dead center is a position where the connecting-rod and crank are in line, so that a thrust or pull along the rod has no tendency to turn the crank.

10. The block 3 in Fig. 2 preserves a constant distance

(parallel to the stroke-line) between crank-pin C and slide 4; but the connecting-rod 3 in Fig. 4, because of its angular swing about the center-line, changes this distance, and modifies the motion of the slide. In Fig. 5 the limits of the stroke-line for the wrist-pin center W, at M and N, are fixed by putting the crank first on one, then on the other, dead center, or by making AM and BN each equal to the rod-length L. If for any crank-position OC the rod-length were measured off horizontally, at CE, it would locate W_0 on MN just as D is located on AB. But when the rod is swung from CE down to its actual position CW, its projected length WD is made less than L; the result is that for any crank-angle (other than 0 deg. or 180 deg.) the point W is farther from M and nearer to N than it would be in harmonic motion. It is evident, further, that if CW is swung about W

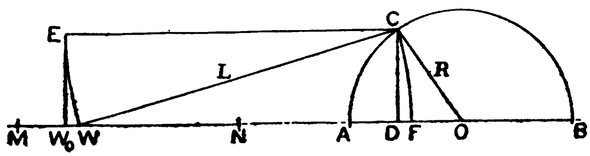

FIG. 5.—Effect of Rod Swing.

down to WF, the point F will have the same location on AB that W has on MN. The difference DF or W_0W shows the effect of rod-angle upon slide-position.

Since a connecting-rod imagined to be of infinite length would not have its angular position changed by the vertical or crosswise movement of the crank-pin, but would always remain parallel to the stroke-line, it would be equivalent in effect to the block 3 in Fig. 2. Hence the motion given to the slide in Fig. 2 is sometimes defined as that belonging to an engine mechanism with infinite connecting-rod.

11. The best graphical method for finding the piston position corresponding to any crank-angle is given in Fig. 6. Having the stroke-limits at M and N, and, with crank at OC, the point W located by striking off the length L from C, proceed as follows:

With M and N as centers and L as radius, draw arcs A_1A_2 and B_1B_2 tangent to the crank-circle. Through C draw EF

parallel to AB, and the position of C on EF will be the same as that of W on MN, or of F on AB in Fig. 5. The simplest geometrical proof of the correctness of the construction is, EC and MW are parallel, ME and WC are equal, therefore MEWC is a parallelogram and EC = MW.

For actual use, nothing outside of the circle AGBH and the tangents A_1A_2, B_1B_2, A_1B_1, and A_2B_2 is drawn: the quadrants are divided into any desired number of parts, and horizontals run out to the nearer tangent arc. By drawing in straight-line tangents, like those on Fig. 3, we can see very clearly the distortion from harmonic motion. Following up the idea of "infinite"

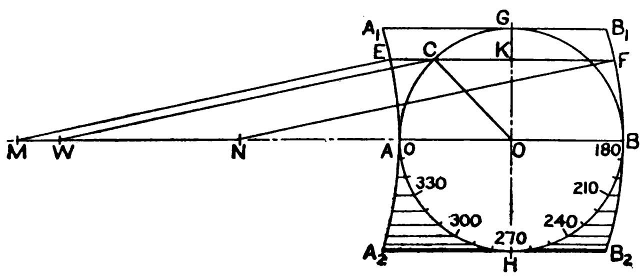

FIG. 6.—Diagram of Piston Positions.

connecting-rod, we may say that these straight lines are arcs of infinite radius.

12. Figs. 3 and 6 are circular diagrams which give the position of the piston for any crank-angle, the crank-circle being the base-line of the figure. To get a straight-line base, we may develop or unroll the circle, and then have a diagram like Fig. 7. The vertical distance AB is the diameter or stroke-length: the ordinates are really successive positions of the line EF (Fig. 3 or Fig. 6) with the point C properly located upon it. Having plotted and drawn the curve for C, we may imagine the line AB to travel along the angle-scale just as the crank turns; then the intersection with the curve will travel on this moving ordinate exactly as does the piston on its stroke-line. On the figure, the

dotted curve is for infinite rod, from Fig. 3, the full-line curve is for the actual mechanism, from Fig. 6.

DRAW PLATE I. SEE APPENDIX,

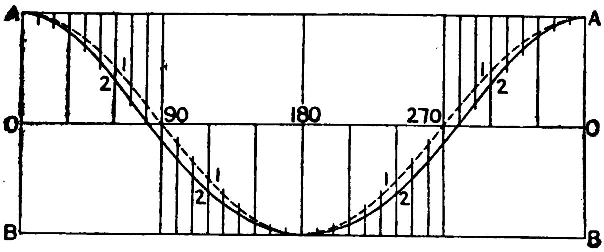

FIG. 7.—The Developed Diagram.

13. The closed curves centered on the radius AO in Fig. 8 constitute a radial or polar diagram of piston position or travel in terms of crank-angle. These polar curves are derived from

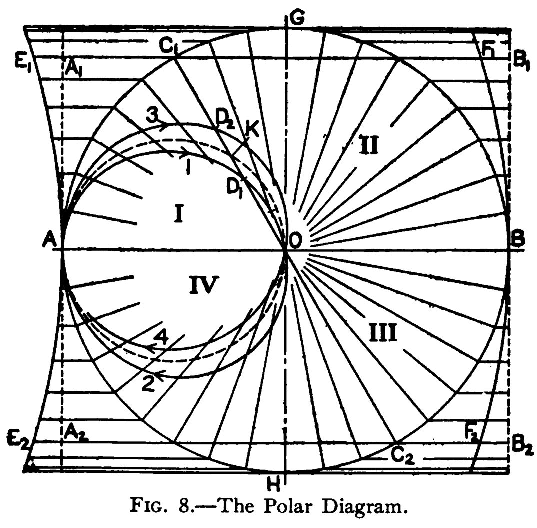

FIG. 8.—The Polar Diagram.

the constructions of Figs. 3 and 6, which are first made. Consider the full-line curves, for the slider-crank mechanism: with the crank at C_1, the travel E_1C_1 is laid off radially inward from C_1

to give a point D_1 on the section of curve which is marked 1, and which belongs to the first quadrant. With the crank at C_2, the distance E_2C_2 is laid off as C_2D_2, and this determines a point on section 3, for the third quadrant. To get the whole diagram, a sufficient number of points are plotted and the curves drawn through them. The arrowheads and numbers show how the successive sections or loops of the complete curve are traced as the crank rotates in the clockwise direction. At A curves 4–1 and 2–3 are tangent, but at B curves 1–2 and 3–4 cross. If we consider the crank as carrying around with it a full diameter of the circle, the intersection D will move back and forth along this rotating diameter just as the point C moves to and fro on the horizontal EF, or as the piston travels back and forth on its stroke-line. All the way round, the distance from the crank-pin point C to the intersection point D shows the distance of the piston from the left end or head end of the stroke.

14. With the crossed slider-crank, or harmonic motion, the two loops of the polar diagram merge in a single curve, around which the intersection point K travels twice during one revolution of the crank. The single polar curve for this case is a circle, as can be shown with the help of Fig. 9. With the crank at OC, the piston displacement from middle of stroke is OD, CDO being a right-angled triangle. If we lay off OE equal to OD, and join AE, we have triangle AEO = triangle CDO; consequently AEO is a right angle and will be inscribed in a semicircle, wherefore E must lie on a circle of which AO is the diameter.

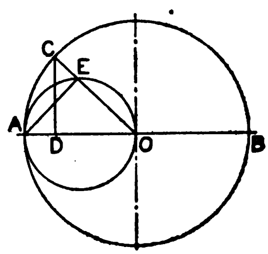

FIG. 9.—Polar Diagram for Harmonic Motion.

DRAW PLATE 2.

15. We have now shown the different types of displacement diagram that are generally available. For the simple slider-crank, there is little, if any, occasion to go beyond the direct circular diagrams, Figs. 3 and 6. But when there is no such simple

relation, and displacements must be found by drawing or plotting the mechanism in successive positions, plotted curves like those in Fig. 7 or Fig. 8 will be the more useful type.

D. The Offset Stroke-Line

16. Sometimes the stroke-line, instead of passing through the center of the crank-circle, is offset to one side, as in Fig. 10. Keeping the line AB, parallel to MN, for the origin of angle-measure, so that the two "strokes" are made while the crank-pin passes over the respective semicircles AGB and BHA, this change makes the strokes quite dissimilar; and the total length of travel becomes greater than the diameter of the crank-circle. The two dead centers are no longer on the same diameter—again defining dead center as the position where crank and connecting-

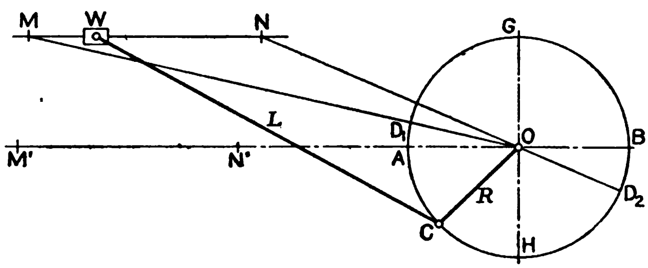

FIG. 10.—Slider-crank with Offset Stroke-line.

rod are in line, and where piston is at extreme of travel. In Fig. 10, the dead centers are shown at D_1 and D_2; on the drawing board, the limiting slide-positions M and N are located by striking off from O the radii $OM=(R+L)$ and $ON=(R-L)$; and the lines MD_1O and NOD_2 are then drawn. Since the arc D_2HD_1 is shorter than D_1GD_2, it follows that with uniform speed of crank rotation the piston must move somewhat faster from N to M than from M to N. Further, the idea suggests itself that a diameter at about the average inclination of OD_1 and OD_2 might be better than the line AB as an origin of angle measurement for this case.

17. The construction for piston position is similar to that with the simple slider-crank, in Fig. 6. From the ends-of-stroke M and N, with rod-length *L* as radius, strike off on Fig. 11 arcs tangent to the crank-circle at D_1 and D_2. Then for any crank position as OC, EC parallel to MN is the piston travel, equal to

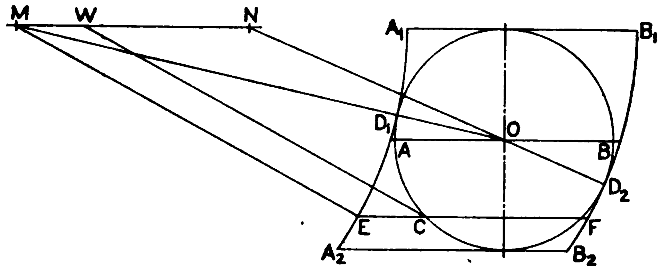

FIG. 11.—Slide-position Diagram.

MW because MWCE is a parallelogram. The total parallel intercept EF is constant, and is equal to the stroke-length MN; the way in which it exceeds the diameter AB is quite evident.

18. In connection with the symmetrical crank-and-slide mechanism, we considered first the simple case of the crossed

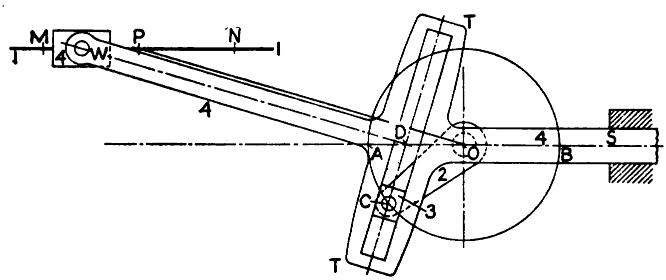

FIG. 12.—Oblique Crossed Slider-crank.

slide, and then introduced the effect of swing of the connecting-rod. It is now of interest to see what would be the movement with offset stroke-line if the angular motion of the connecting-rod were eliminated. This condition is realized in Fig. 12, where the slot TT is perpendicular to the rod-line DW. The length DW

is the same as CW or L on Fig. 10 or Fig. 11, and this line has parallel motion, with one end always on AB, the other on MN. The middle of the stroke is located on MN by striking off L from O as OP; and this point will serve as the best reference position in plotting the motion of the slide in the mechanism of Fig. 10. As compared with Fig. 2, we now change from perpendicular to oblique projection of C upon AB, at D. The geometrical relations in this oblique slider-crank are set forth in Fig. 13, where KL is the line of rod-slant, and the limit-lines KM and LN are tangent at K and L. The piston displacement is now EC or A′D. The line KL meets the suggestion at the end of paragraph 16 as to a better origin of angle for the offset slider-crank than the diameter AB.

10. There is a simple relation between the movement of the

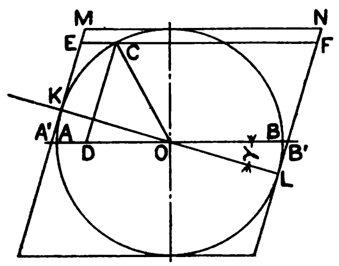

FIG. 13.—Diagram for Oblique Crossed Slide.

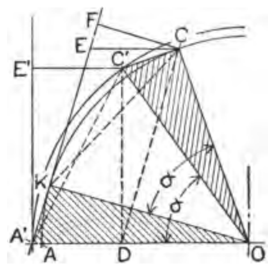

FIG. 14.—Geometrical Relations

point D in Fig. 13 and in Fig. 1, which may be established as follows:—

Take from Fig. 13 the triangle KOA′, and in Fig. 14 rotate it about the center O, through any angle α, to a position COC′. Take also the inclined limit-line KE from Fig. 13, and from A′, where EK cuts the diameter AB (or the radius AO), erect a vertical limit-line A′E′. Draw the horizontals CE and C′E′, parallel to AO; it can be shown that these lines are equal, and from this equality results the relation sought. To prove that EC and E′C′ are equal, draw CF perpendicular to A′E; then considering the quadilaterals A′E′C′O and KFCO, we see first that the triangles

A′OC′ and KOC are similar, and second that the four angles are respectively equal, because those at A′, E′, K, and F are right angles; consequently, these figures are similar, and we have

$$E'C':FC::A'O:KO.$$

But the triangles EFC and A′OK are similar, wherefore

$$EC:FC::A'O:KO;$$

whence $E'C' = EC$, and if lines are drawn from C′ and C, parallel respectively to E′A′ and to EK, they will meet in the point D.

Now EC is the piston dipslacement for Fig. 12, by the method of Fig. 13; and E′C′ is the similar displacement for a plain crossed slider-crank with a crank of the radius OA′. In other words, the motion given by the oblique arrangement in Fig. 13 is exactly like that of the mechanism in Fig. 2, but is timed a little differently (by the amount of the constant angle KOA) and is, for the same radius of actual crank, of a little greater magnitude (in the ratio of OA′ to OA).

Draw Plates 3 and 4.

20. As illustrating the effect of a difference in point of view, we may note that when the mechanism of Fig. 10 is applied in an engine, the crank is said to be offset, not the slide or cylinder. In the engine, the cylinder, as the place of power development, is the more prominent member of the machine, or at least the one to be first considered; and the crank-shaft is therefore located from the center-line of the cylinder. In a kinematic study, the crank is the determining member, hence the reversal of procedure.

E. Distorted Harmonic Motion

21. In the straight slider-crank, the distortion from harmonic motion caused by the swing of the connecting-rod is an incidental effect, unavoidable and not particularly desirable. We now come to a class of mechanisms which, primarily driven by a rotating

crank, are intended to produce a marked distortion from simple harmonic motion. The most prominent example is the valve-gear of the Corliss engine, of which a typical example is outlined in Fig. 15.

Motion is initiated by the short crank or eccentric OE, rotating about O; through the eccentric-rod EF, the rocker-arm AB, and the reach-rod BC, it is imparted to the oscillating wrist-plate W, with its center of turning at G. The rods EF and BC

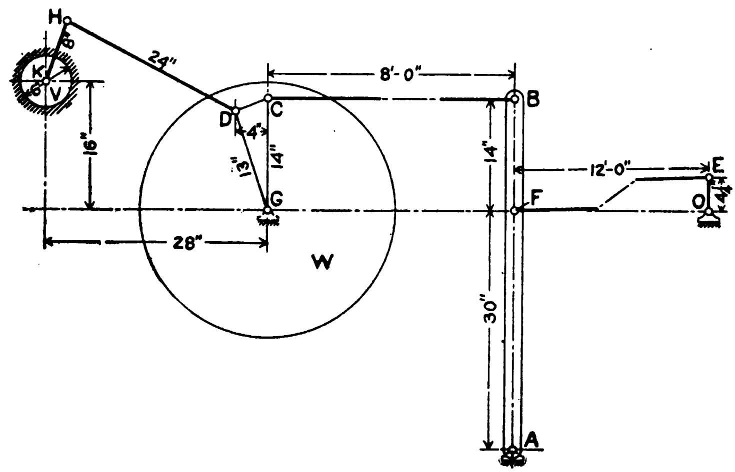

FIG. 15.—General Outline of Corliss Valve-gear.

are so long that the effect of their angular swing is practically negligeable, and we may assume that the point C has harmonic motion horizontally. Then the mechanism with which we are really concerned is made up of the wrist-plate (reducible to a triangle CGD), the valve-rod DH, and the valve-arm HK. What we finally wish to determine and plot is the movement of a point on the circumference of the valve V, along its curved path of motion.

22. In Fig. 16 the first thing is to determine the movement of the point C on the wrist-plate. On the rocker-arm, the travel of point B is greater than that of F in the ratio of AB to AF;

using the dimensions on Fig. 15, and noting that E has a horizontal movement through 8.5 in., we find the travel of B (and of C) to be

$$8.5 \times \frac{44}{30} = \frac{187}{15} = 12\frac{7}{15}, \text{ practically } 12.5 \text{ in.}$$

This then is the diameter of the crank-circle for the equivalent direct drive; and if we draw this circle on AB, Fig. 16, divide it as desired, and project the division-points vertically to the arc EE

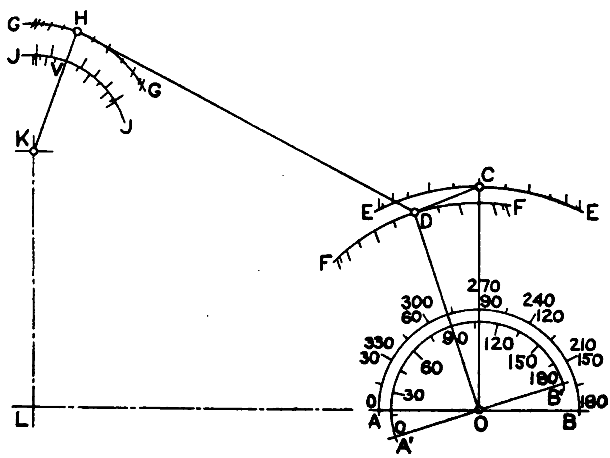

FIG. 16.—Determination of Valve Movement.

which is the path of C, we have C located for each of the chosen crank (or eccentric) positions.

To pass from driven point C to driving point D, we might project the points on arc EE radially inward to arc FF, then rotate the resulting scale of positions till it centered itself on D. It is simpler, however, to reduce the equivalent direct eccentric-circle in the ratio of OD to OC, center it on the line OD, and, after dividing, project parallel to OD upon the arc FF. The diameter

A′B′ is $\frac{187}{15} \times \frac{13}{14} = 11.55$ in.

23. Having located a set of positions for D, it is an easy matter to strike off DH from each point on the arc FF and get corresponding points on the arc GG. This gives the movement of H; for that of V (which may be thought of as a point on either the valve-arm HK, in line with the surface of the valve, or on the valve-profile itself) we project radially inward, getting the position-scale marked on the inner side of the arc JJ. Finally, a distance-scale, stepped off along JJ with intervals of say one-half inch, and marked on the outer side of JJ, helps to develop or rectify the curvilinear displacements of V along JJ.

It will be noted that the mechanism is drawn in mid-position, with the eccentric-radius OE, Fig. 15, standing at 90 degrees.

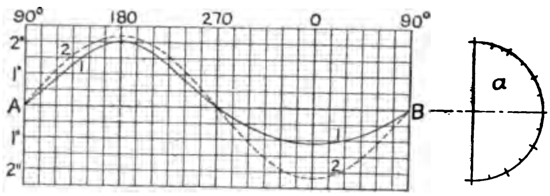

Fig. 17.—Developed Diagram of Valve Movement.

This middle of stroke, instead of the end of the stroke, is the proper reference position for a valve-gear. Further, in Fig. 16 the radius KV is larger in proportion than on Fig. 15: if it be convenient, for instance, to lay out the mechanism half-size on Fig. 16, it is well to draw the valve full-size, and thus get full-size displacements directly.

24. The most useful diagram of the results obtained in Fig. 16 is of the form of Fig. 7. It is drawn in Fig. 17, where the central base-line AB corresponds to the "mid-position" at V, and the vertical divisions (distances between ruled lines) to the scale on the outer side of the arc JJ, Fig. 16. In the plot, travel of V toward the right is shown by an upward ordinate, travel toward the left by a downward ordinate. The result is the full-line curve 1; and the greater travel toward the right than toward the left is strikingly shown by this curve. The wider movement

is used to open the valve, the short movement comes while the valve is closed.

The harmonic motion curve 2 is drawn for the sake of comparing this mechanism with the most nearly equivalent plain crossed slide driving a flat valve or slide. The diameter of the eccentric-circle at *a*, Fig. 17, is fixed by taking account of all the ratios of lever-arms in Fig. 15; having found A′B′, Fig. 16, by using two ratios, we multiply it further by KV/KH, and get $11.55 \times \frac{3}{8} = 4.34$ in.

The movement determined in Fig. 16 can also be very well represented by a polar diagram like Fig. 8.

DRAW PLATE 5.

F. Valve Diagrams

25. The valve-gear of a slide-valve engine, sketched in Fig. 18, and consisting of eccentric 2, eccentric-rod 3, and valve-slide 4 as moving parts, is the same in kinematic form as the main

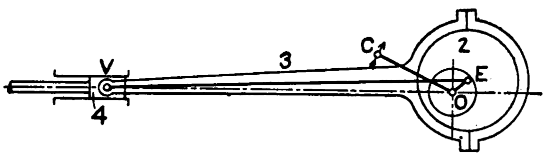

FIG. 18.—Outline of Common Valve-gear.

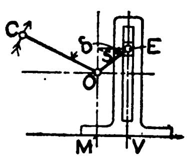

FIG. 19.—Equivalent Crossed Slide.

mechanism of the engine: but the rod is here so long relative to its crank that the effect of its angular swing may be neglected in most cases, and we have the simple condition of harmonic or crossed-slide motion to consider. The mechanism may be reduced to the elementary form in Fig. 19, and the thing sought is a diagram which will give the displacement of the valve from mid-position, MV or SE, for any position of the crank OC.

The combined layout of the two mechanisms in Fig. 20 helps to make clear the requirements of the problem, and illustrates

a convention to be adopted as to direction of valve movement. In order to open the port at the left end of the cylinder and admit steam for the forward stroke of the piston, the valve must move toward the right from mid-position. For this reason, we consider right-hand travel as plus, left-hand travel as minus, in fixing the directional meaning of the ordinates of valve diagrams.

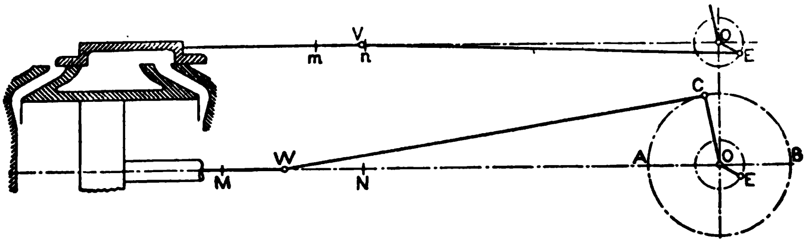

FIG. 20.—Engine and Valve-gear Mechanism.

26. The simplest diagram, perhaps, is that of Reuleaux, derived in Figs. 21 and 22. Fig. 21 merely reproduces the geometrical part of Fig. 19, showing the rigid crank-and-eccentric

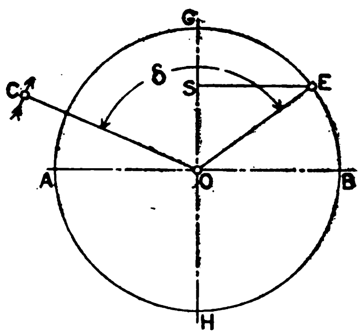

FIG. 21.—Valve-position Diagram.

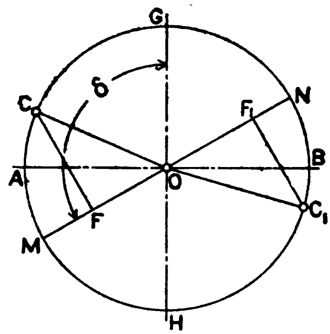

FIG. 22.—The Reuleaux Diagram.

COE, with the angle δ between the arms. For any eccentric position as OE, the valve displacement is, directly, the distance SE; but if a crank position is given, as at OC, we must measure forward the angle δ and thus locate OE, in order to get SE. To eliminate this angle-measurement, take the figure OES on GH

and turn it backward about O through the distance δ, bringing OE into coincidence with the crank at OC, and the reference-line GH (of Fig. 21) to the position MN, which is constant for a given value of δ. Now, for any crank position OC, the valve travel (or distance from mid-position) is given by the line CF, perpendicular to MN. If FC is measured upward and to the left, C being on the arc MGN, the travel is plus or toward the right; but if C is anywhere on the arc NHM, as at C_1, the valve is to the left of mid-position and the travel F_1C_1 is minus.

27. The Zeuner valve diagram is based on the polar diagram in Fig. 8, taking the case for harmonic motion; but the circle is placed on the right-hand side of the center-line, in Fig. 23, because the intercept on the radius itself is now to show travel toward the

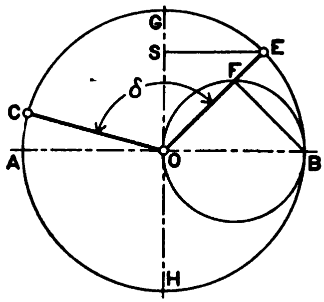

FIG. 23.—Direct Polar Diagram.

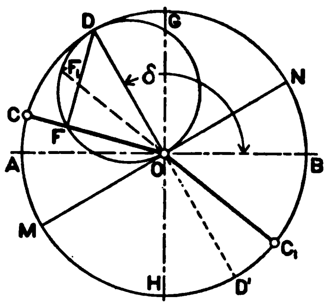

FIG. 24.—The Zeuner Diagram.

right, while an intercept on the extension of OE backward, through O, is to show left-hand or negative travel. Again, as in Fig. 22, we change from a diagram in terms of eccentric position to one in terms of crank position, by rotating backward about O, through the angle δ, the figure made up of the circle OFB and the radius OE. Then, in Fig. 24, the circle takes a constant position on the diameter OD, and the eccentric is brought into coincidence with the crank. The intercept OF, cut from the crank by the valve-circle, now measures travel, which is to the right if F is between C and O, but to the left if the radius must be produced, as C_1O to F_1.

28. If the eccentric OE be placed on its plus dead center—

which is away from the cylinder and opposite to the head-end crank dead center which is taken as the zero of crank-angle—the crank will be perpendicular to the base-line MN of the Reuleaux diagram, and will lie along the diameter OD of the Zeuner valve-circle; and in each case it will show the greatest plus value of the

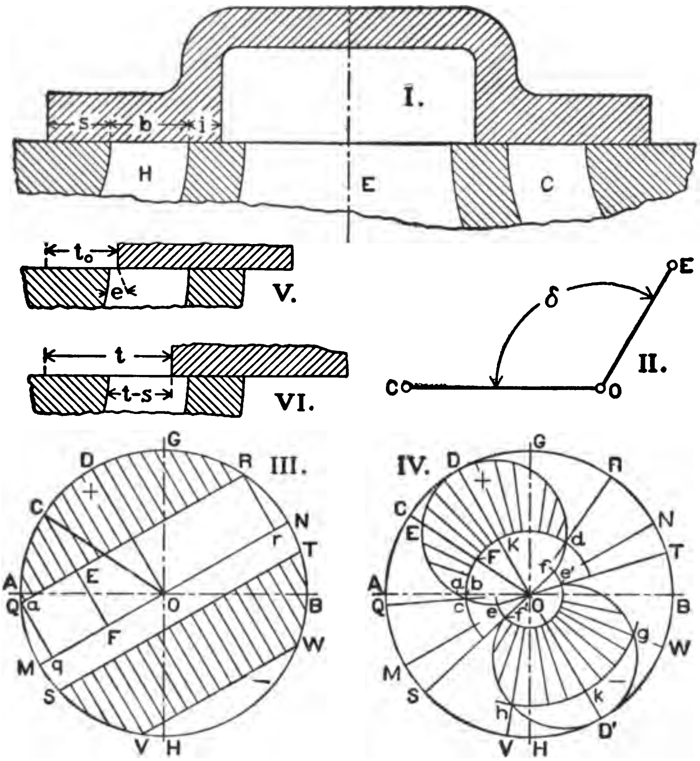

FIG. 25.—Complete Valve Diagrams.

travel *t*. In these facts lie sufficiently definite rules for the drawing of either diagram.

The only essential difference between these diagrams and Figs. 3 and 8 is found in the fact that we are now showing movement due to one crank in terms of the position of another crank which makes a constant angle with the first. In using the valve diagram,

no attention need be paid to the actual position of the eccentric. In Fig. 22, just as C moves in and out from F, so the valve moves in and out from its reference position; in Fig. 24, the movement of the point F on a rotating diameter through CO shows the movement of the valve on its seat, with C always representing the plus end of the valve-stroke. Not only does FC or OF show valve position, but we can see which way the valve is moving at any instant by noting whether this distance will increase or diminish as the crank advances.

29. The object in completely representing valve movement by means of a diagram is to be able to show just how the valve acts in opening and closing the steam port for the admission and exhaust of steam. In Fig. 25, sketch I shows a common slide-valve at mid-position on the valve-seat. The controlling dimensions, besides r and δ as represented at II, are

s = outside lap or steam lap;
i = inside lap or exhaust lap.

The breadth b of the port is of secondary importance in the present connection. Complete valve diagrams for the left port, or for the head end of the cylinder, are given at III and IV.

When the crank is at OM—in IV this line is tangent to the valve-circle, so as to have a zero-intercept—the valve is in mid-position. As the crank advances, the valve moves toward the right, the t-ordinate being positive and increasing in both figures: when C gets to Q, where $t = \mathrm{Qq} = \mathrm{cO} = s$, the valve-edge and port-edge are just in line, or the port is just beginning to open. For the crank on dead center, the valve takes the position shown at V; the travel is t_0 and the port is open by the small amount e, which is called the lead. When the crank is at any position OC, to which VI corresponds, we have $t = \mathrm{CF} = \mathrm{EO}$: and it is evident that, in general, the port-opening is equal to $(t-s)$. In order to make a graphical subtraction of s from t, we draw in III the lap-line QR parallel to MN at the distance s; and in IV, draw the lap-circle cKd, with s as radius. Then the segment QDR and the cresent cDd are identical diagrams of port-opening. We

see that admission begins—or, we "have admission"—at Q, maximum opening is at D, while cut-off takes place at R. It is evident that the determining of admission and cut-off is simply a matter of finding crank positions for which the valve is at a certain distance from mid-position.

30. After the crank passes R, the valve keeps on moving back from the right—as is shown by a plus but decreasing t—until it gets to mid-position again when the crank is at ON: then it goes toward the left, and soon opens the exhaust-port, this occurring when $t = -i$. The beginning and end of exhaust, or "release" and "compression," as also the port-opening during exhaust, are found by drawing the exhaust lap-line TS or the inside lap-circle fOe. In IV, instead of using only the plus valve-circle on OD, we save overlapping by drawing another valve-circle on OD′, for which the direct intercept shows left-hand or minus travel. This is convenient but not necessary, for it is evident that OT is determined equally well by either intersection, e or e′. In finding release and compression from the Zeuner diagram, the beginner is likely to confuse the intersections of valve-circle and inside lap-circle, especially when, as is usual, only the one valve-circle is drawn. Keep clearly in mind, not only that the valve must be at a certain distance for one of these events, but also to which side it must be, and which way it must be moving. Thus, with the positive valve-circle alone, if we were to draw a crank-line from O through f for the release-position, we should make a mistake: for while the valve is at the distance i, it is toward the right; whereas it should be to the left and moving to the left, as is the case when the crank is at eOT. For these short-lap measurements the Reuleaux diagram is clearer and more accurate than the Zeuner.

31. On the exhaust side of this valve there is over-travel; for if we measure off the port-width b, and draw VW parallel to ST, and the circle hkg at the distance b from the lap-circle, we see that the valve travels more than enough fully to open the port. Sometimes there is a slight over-travel on the steam side: but more frequently—and most of the time in single-valve gears

with variable cut-off—the maximum opening for admission is much less than the width of the port.

For the other port, or the other end of the cylinder, the events and conditions are diametrically opposite to those shown, with a symmetrical valve: if the laps are not equal, they must be drawn in, and the required intersections found. Generally, both sets of lap-lines should be drawn on a Reuleaux diagram, dotting those for the crank end. But in the Zeuner diagram, we usually

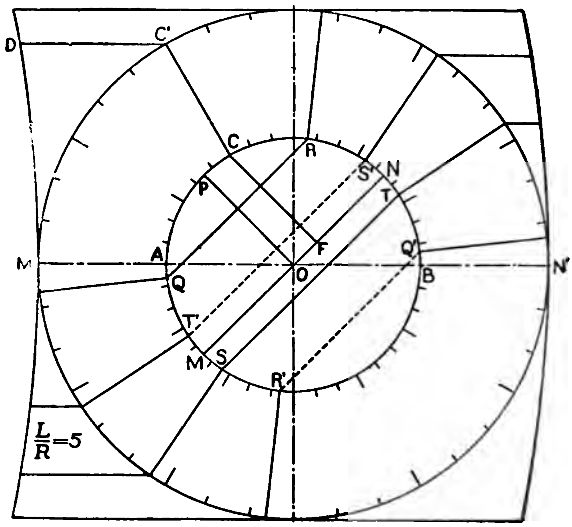

Fig. 26.—Piston and Valve Diagrams.

draw only the one valve-circle; and, for equal laps, the same circle serves for both ports.

32. Having established simple methods for finding the relation between the positions of the valve and of the crank, our next step is to extend these to the valve and piston. The primary, determining diagrams are shown in Fig. 26, where the valve-diagram (of either form) is combined with the piston-position diagram from Fig. 6: then DC′ and CF are simultaneous determinations. The distortion from symmetrical steam-distribution caused by

the action of the connecting-rod, notably the inequality in the cut-offs,* is well brought out by this figure; but can be rather more clearly seen on the derived diagram given as Fig. 27, where the valve-travel is plotted on the piston stroke-line as a base. The abscissa in Fig. 27 is DC′ from Fig. 26, the ordinate is FC.

The curve got by this method is elliptical in form, and with harmonic motion for the piston as well as the valve it is a true

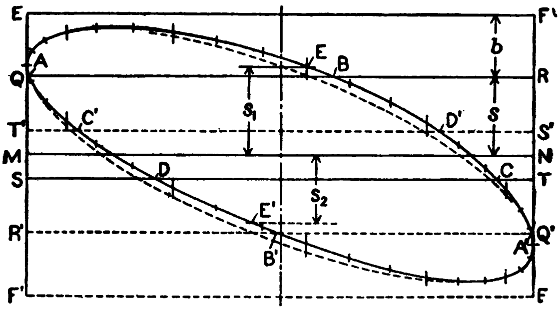

FIG. 27.—Valve Movement with Reference to Piston.

ellipse. The effect of the connecting-rod is here shown by dotting in parts of the simpler curve. The lap-lines are now drawn parallel to MN: and the four events, admission, cut-off, release, and compression, are located by the intersections marked A, B, C, and D, respectively. Dotted lines and primed letters are for the crank end of the cylinder. The points E and E′ show cut-offs which are "equalized," or made to take place at the same distance from the respective beginnings of stroke, while s_1 and s_2 are the resulting unequal steam laps. The movement produced by valve-gears of the type of Fig. 15 can be very effectively shown by means of a diagram like Fig. 27.

DRAW PLATE 6.

* This means, inequality in the distances travelled by the piston, from the beginning of either stroke to the position where the valve closes.

G. Velocity Relations

33. In all the examples thus far considered, and in the great majority of machines, the motions of the members of a mechanism are in a plane or in parallel planes. Typical examples of the several kinds of coplanar constrained motion are found in the one machine outlined in Fig. 28, which shows the mechanism of the ordinary locomotive engine. The different kinds of motion here illustrated are as follows:

a. Straight-line translation, or parallel motion, of the slide WD (crosshead to piston); all points on this piece move in straight parallel paths with the same velocity.

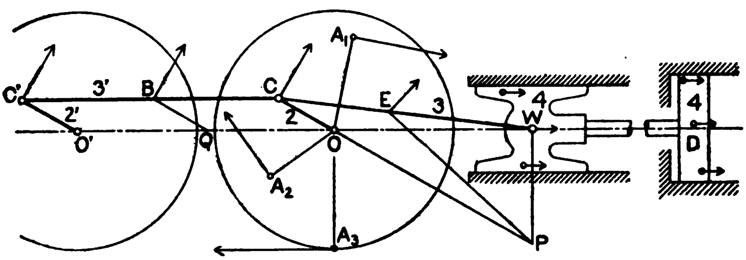

FIG. 28.—Various Kinds of Motion.

b. Curved-line translation of the coupling-rod CC′; all points have the same velocity (equal and parallel), but the point-paths are curved, so that the common velocity is continually changing in direction.

c. Simple rotation, about a fixed center, by the crank and wheel; the velocity of any point A is perpendicular and proportional to its radius OA from the center.

d. Rotation about a changing or "instantaneous" center; the connecting-rod WC is, at the instant, rotating about a center P which lies at the intersection of the lines WP and CP, drawn perpendicular to the known path-directions (or velocity-lines) of two points on the rod. This is the general case, covering all motion of bodies in a plane.

34. The idea that any motion of a body in a plane can be, and is, produced by a rotation about a center, is a fundamental concept of the greatest importance in kinematics. Rotation about a fixed center is nothing but the simplest case of this action. In general the center must change, with a smooth, continuous motion of its own; but this lack of permanence does not at all affect the general relation, that for any point on a rigid body the velocity is perpendicular and proportional to the radius from the center of rotation.

That rotation is the universal type of motion is so fundamental a fact that the only way to " prove " it is to find, by repeated trial, that it meets every case: and to this end, actual use and application of the idea is the most convincing argument. Even the translational motions in cases *a* and *b* of Fig. 28 come under this scheme as special cases. With straight-line translation, we simply say that the center is at an infinite distance away, on a line (any line) perpendicular to the line of motion. Case *b* is the most peculiar, from this point of view, that can be found: here the center is at an infinite distance on a rotating radius, and is therefore moving with infinite velocity. This non-rectilinear translation is the one case in which the idea of a single instantaneous center for the body is of no practical use.

35. Consider Fig. 29, where the engine mechanism is shown in the more usual position. It is very evident that if the rod as a whole is in rotary motion, with the requirement that direction of point-movement be at right-angles to radius, the intersection at P must be the center. The point C, considered as the end of the radius OC, is rotating about O; it may seem strange to say that the coincident point C on the rod is rotating about P; but we must remember that at the instant C is simply moving in a path-element perpendicular to OP. It is because the path is turning toward O that this center can be fixed while P must move.

Now just as C is at once a point on crank 2 and on rod 3, and W a point on rod 3 and on slide 4, so P is to be thought of as the common position of two points, one definitely located on a plane attached to the frame 1, the other on a plane attached to the rod 3. Each of these points lies on a locus of centers, a curve called

a centrode. As the two bodies go through their relative movement, the centrodes roll on each other.

In Fig. 30 the centrodes for the motion of connecting-rod relative to frame are partly laid out. The plane of the paper being attached to piece 1, the successive determinations of P as on Fig. 29 give centrode 1 directly. The dotted outline indicates the process of plotting the curve on 3. Since centrode 3 is to be shown for rod at WC, we take the triangle P′W′C′ and set it on the rod as P″WC, thus getting the point P″ on curve 3.

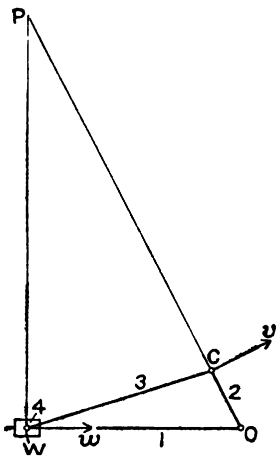

FIG. 29.—Instantaneous Center, Engine Mechanism.

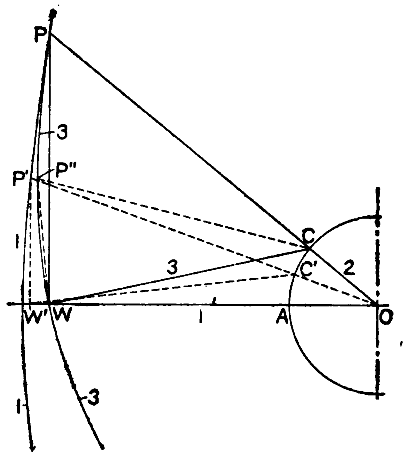

FIG. 30.—Example of Centrode.

These centrode curves are of considerable interest, but are of little, if any, practical use so far as the objects of the present course are concerned. There are, however, certain properties of the instantaneous center which are exceedingly useful in determining velocity relations.

36. In connection with Fig. 29, it has been made clear that P is to be considered, not as a point fixed in space, but as a point definitely located on the plane of frame 1. In Fig. 28, where the machine as a whole has a progressive motion which we have not been considering, this statement of condition is even more important. If we take frame 1 as the reference piece, with "zero"

velocity, then P on rod 3 is the one point on the plane of that piece which has the same velocity as 1. This leads up to the general definition of the instantaneous center, *That as concerns the relative motion of two machine pieces, this center is the common position of two points that have identical velocities or are moving together—the points being on or attached to the respective pieces.*

37. Fig. 31 serves to illustrate a usual notation for instantaneous centers, and also to demonstrate the very important theorem of three centers. The notation consists in joining the numbers of the two links concerned: thus the center for links 2 and 3 is 23—read "two-three" not "twenty-three." The theorem of three centers is, *That any three links of a mechanism must have their three relative instantaneous centers on a straight line.*

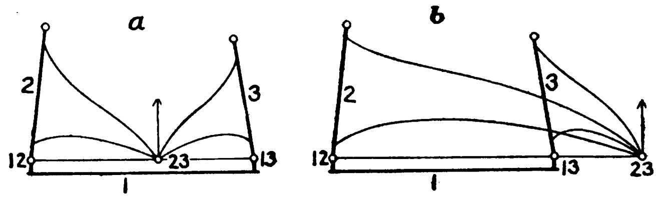

FIG. 31.—Theorem of Three Centers.

In Fig. 31, consider the links 2 and 3, which turn on 1 at 12 and 13. Any point on the plane of link 2 will have its velocity perpendicular to a radius from the center 12; any point on 3, to a radius from 13; and points with the same direction of velocity can be got only by taking them on radii which lie along the same line, the line 12–13. Without fuller data than appear on the figure, we can do no more than say that the point of identical velocities will be somewhere on this line; it will lie between 12 and 13 if links 2 and 3 have opposite directions of rotation, but if the directions are the same, 23 will fall beyond one or the other of the primary centers.

38. In Fig. 32 this principle is applied in finding the point of common velocity for the piston-slide 4 and the crank 2. Considering links 2, 1, and 4, we have that 24 must lie on the line 12–14, which is the vertical at 12, since 14 is at infinity. Again, con-

sidering 2, 3, and 4, we have 24 on the line 23–24, or the center-line of the rod. The intersection fixes 24 where it satisfies both relations.

Sketch *b* is intended to emphasize the properties of this center 24 or B, more effectively than can be done with the rather fantastic scheme for indicating " attachment " that is used at *a* and in Fig. 31. The slide is placed back of the crank, and it is made clear that the two pieces are moving together as if, at and for the instant, they were joined by a pin at B. For the crank, all points on an instantaneously vertical radius have horizontal path or velocity;

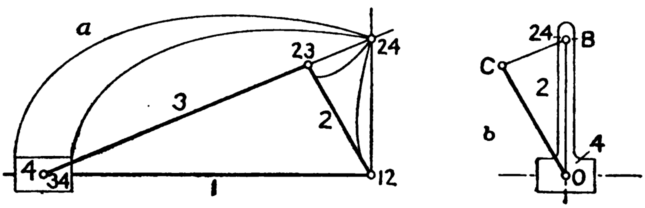

FIG. 32.—Instantaneous Center and Effective Lever-arm.

and this velocity can have, as a point changes position along the radial line, any value from plus infinity, through zero, to minus infinity. At B is the particular point which has the same velocity as slide 4, and OB may very well be called the ***effective radius*** with which the crank drives the slide. This idea of an effective lever-arm or driving-arm for velocity will be found very useful; but before applying it to other mechanisms, we will consider more fully the action of the slider-crank.

39. Let v be the linear velocity of the crank-pin and w the velocity of the wrist-pin or the slide; then the relation shown in Fig. 32*b*, when expressed by a proportion, is

$$w:v::\mathrm{OB}:\mathrm{OC}.$$

In Fig. 33 are illustrated two ways of proving that the second ratio is equal to the first, and thus establishing by other lines of approach the geometrical relation embodied in the triangle OCB.

First, use the instantaneous center P of rod on frame: of necessity,

$$w : v :: \mathrm{PW} : \mathrm{PC};$$

but triangles PCW and OCB are similar, therefore

$$\mathrm{OB} : \mathrm{OC} = \mathrm{PW} : \mathrm{PC} = w : v.$$

Next, take the velocity v at C, and resolve it into components, u_1 or CF at right-angles to WC and r or CE along WC; likewise, at W, resolve w into u_2 or WK across and r or WH along the rod.

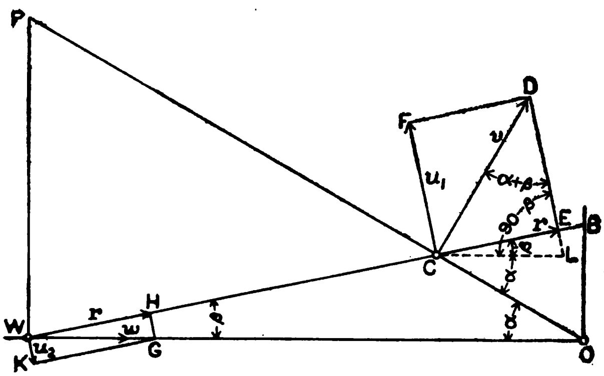

FIG. 33.—Velocity Relations.

Since the line WC is unchangeable in length, the components r must be the same in value. Then the triangle WHG may be moved to CEL, and the triangle CDL will give a simple relation between v and w, determined by the fact that DL is perpendicular to WC. Now triangle OCB is similar to CDL (sides perpendicular in pairs), consequently

$$\mathrm{OB} : \mathrm{OC} = \mathrm{CL} : \mathrm{CD} = w : v.$$

This velocity-by-resolution method is not so simple as that of instantaneous centers, and is introduced here largely to show how the same result can be attained in several ways. In some lines of velocity analysis, which we shall not now take up, this method is very important and useful.

H. Velocity Diagrams

40. Continuing the discussion of the slider-crank mechanism, we have in Fig. 34 the simplest diagram for showing velocity of piston in terms of crank position. It is based on the relation, which has just been so fully demonstrated, that the center-line of the connecting-rod cuts from the vertical GH an intercept which bears to the piston velocity the same ratio that the radius of the crank does to the crank-pin velocity. We assume the latter velocity to be constant, as it practically is in any engine with a sufficient flywheel; and further, we choose our velocity scale so that the length OC will just be equal to this v.

Directly, then, slide velocity is given by the length OD on Fig. 34. If now we draw OL parallel to the rod CW, it will cut from

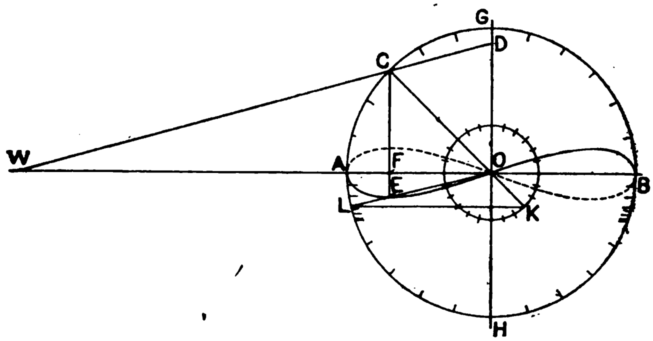

FIG. 34.—Construction for Piston Velocity.

the vertical CF a length CE which is equal to DO. The locus of E is of a figure-of-eight form, the full-line part for the forward stroke, the dotted part for the return stroke. With "infinite" connecting-rod, CD would be horizontal and CF would give the velocity w, instead of CE.

Referring to Fig. 33, we see that if in either triangle CDL or OCB we apply the proportionality of sides to sine or opposite angles, we get

$$w:v::\sin(\alpha+\beta):\sin(90-\beta),$$

or

$$w = v\frac{\sin(\alpha+\beta)}{\cos\beta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

With infinite connecting-rod, the rod-angle β would be zero, and we should have

$$w = v \sin\alpha, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

as is also apparent for CF in Fig. 34.

41. In drawing the E-point curve, centered on AB in Fig. 34, the essential thing is the line of rod-slant, OE or OL. In that figure is shown a reduced construction for this line, kept wholly within the crank-circle, the triangle LOK being a miniature of

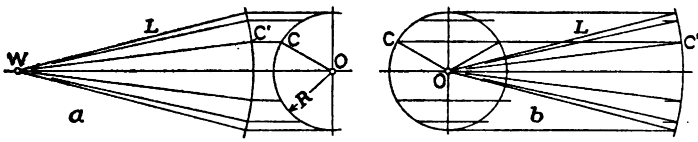

FIG. 35.—Rod-angle Construction.

the mechanism-outline WCO. The principle of this reduced rod-angle diagram is more fully illustrated in Fig. 35. At *a*, the rod-length *L* (to any desired scale) is struck from a center W, giving the arc through C′; from any center O on the line of stroke, a crank-circle is drawn with radius *R* (to the same scale), and crank-pin positions like C are projected over to C′. The purpose of this figure is to emphasize the fact that the crosswise movement of C, or the distance of C from the line WO, is what determines rod-slant; and it is evident that both radii, *L* and *R*, may be struck from the same center if desirable, as at *b*. In Fig. 34, the main radius OC is taken to represent rod-length, at OL; and the small circle is then drawn in the proper ratio, to represent the crank-circle to this same reduced scale. Crank-pin positions on the large circle are carried radially inward, resulting division points on the small circle are projected over to the large one near A or B, and lines like OL are drawn as needed.

Fig. 36 shows how the same device may be used with the offset

stroke-line. Diagram *a* is self-evident; at *b*, point O represents the center of the large circle in a diagram like Fig. 34, the little circle being put here so as to use most readily the angle-divisions on the big circle. The rod-length must now be struck from the offset point Q, giving the arc AB as the locus of the point C′, while DE is a part of the main circle.

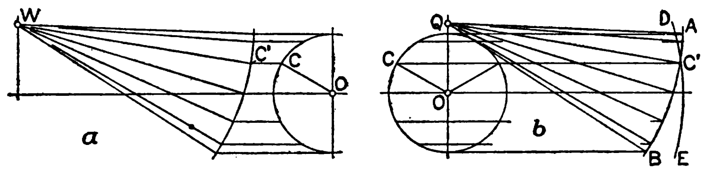

FIG. 36.—Rod Angle With Offset Stroke-line.

It can easily be seen that the construction for piston-velocity in Fig. 34—which consists in rod-slant line OE cutting off the length CE from the vertical CF—applies equally with offset stroke-line.

42. Besides the diagram of piston velocity in terms of crank position, it is desirable to have one in terms of piston position.

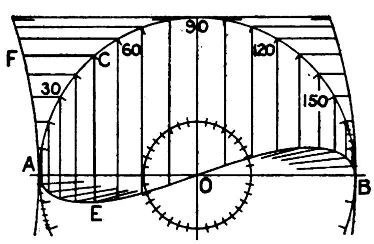

FIG. 37.—Piston Velocity and Position.

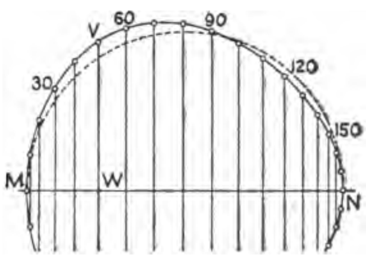

FIG. 38.—Velocity Diagram on Stroke-line.

To get this, we first combine Figs. 6 and 34, in Fig. 37, getting piston displacement in FC and piston velocity in CE. The same radius is taken to represent both crank-arm R and crank-pin velocity v; if desirable, different radii might be used, after the manner of Fig. 26. In Fig. 38, travel FC is laid off as MW, velocity EC as WV. The full-line curve through V is the result,

while the dotted circle is the similar diagram for the case of harmonic motion.

A third possibility, which may sometimes be of use, is a polar diagram of piston velocity, analogous to Fig. 8. With harmonic motion, this is a circle on a vertical radius, on OG instead of OA, Fig. 8: for the slider-crank mechanism, the curve must be plotted from Fig. 34. In this velocity diagram, we would separate the curves for the two strokes, using for the return stroke a circle on OH and a curve plotted below AB.

DRAW PLATE 7.

I. Quick-Return Motions

43. Taking the outline of the slider-crank mechanism in Fig. 39*a*, we define its essential elements, or the "links" in this kinematic "chain," as follows:

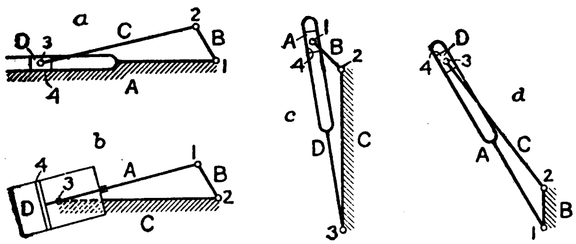

FIG. 39.—Inversions of Slider-crank.

A and D carry each one turning and one sliding element;
B and C each two turning elements;
C is several times as long as B;

A joins B, the shorter, D joins C, the longer of the two links with two turning elements each.

These definitions are a help in keeping track of the links in the inversions of this mechanism shown at *b*, *c*, and *d*. At *b* and *c*, link C, the connecting-rod in the ordinary engine, is the piece held fast as the frame of the machine; sketch *b* shows the oscillating engine (see also Problem 1), sketch *c*, the crank and slotted lever

mechanism, which is sometimes used as a quick-return drive in certain machine tools. These two arrangements differ chiefly in the form of the sliding pair, joint 4. Comparing *b* with *a*, we see an inversion of the pair, in the interchange of solid and hollow parts; between *b* and *c* there is a decided difference in the form of the links, but in neither case is there any change in kinematic value or effect.

Sketch *d* shows link B, ordinarily the crank, as the fixed member. This mechanism has a number of applications, but the only one that we shall consider here is the Whitworth quick-return motion. A fourth possibility is the holding fast of link D, which makes the crank link B the floating member of the mechanism (without contact with the frame). This scheme has no useful application of any importance, and we shall give it no further attention.

44. In Fig. 40, mechanism *c* of Fig. 39 is applied in a machine, driving slide 6 through rod 5. We return to the regular kinematic notation of Figs. 31 and 32 (using letters also at a number of important centers). Crank 2 rotates at a uniform speed, driving lever 4 in an oscillating motion which is more rapid when the crank-pin is in the lower part of its circle than when it is in the upper part. The important derived centers are 24 and 46, which are obtained as follows:

$$24\begin{cases}23\text{–}34\\12\text{–}14\end{cases}\qquad\qquad 46\begin{cases}14\text{–}16\\45\text{–}56\end{cases}$$

Now 24 is a point which moves with the same velocity on crank 2 and on lever 4, and 46 is a similar point for lever 4 and slide 6: or, OF is the effective arm with which crank drives lever, and AG is the effective arm with which lever drives slide. Letting OC represent velocity of point C, OF will show velocity of F or 24. On the proportion diagram at II, which is separated from I to avoid confusion of lines, draw FH perpendicular to AF and equal to FO; then with GK parallel to FH draw from A, the center of rotation of the lever, a line AHK to fix the length of GK as the velocity of G or 46 on the lever; and this GK will also be the

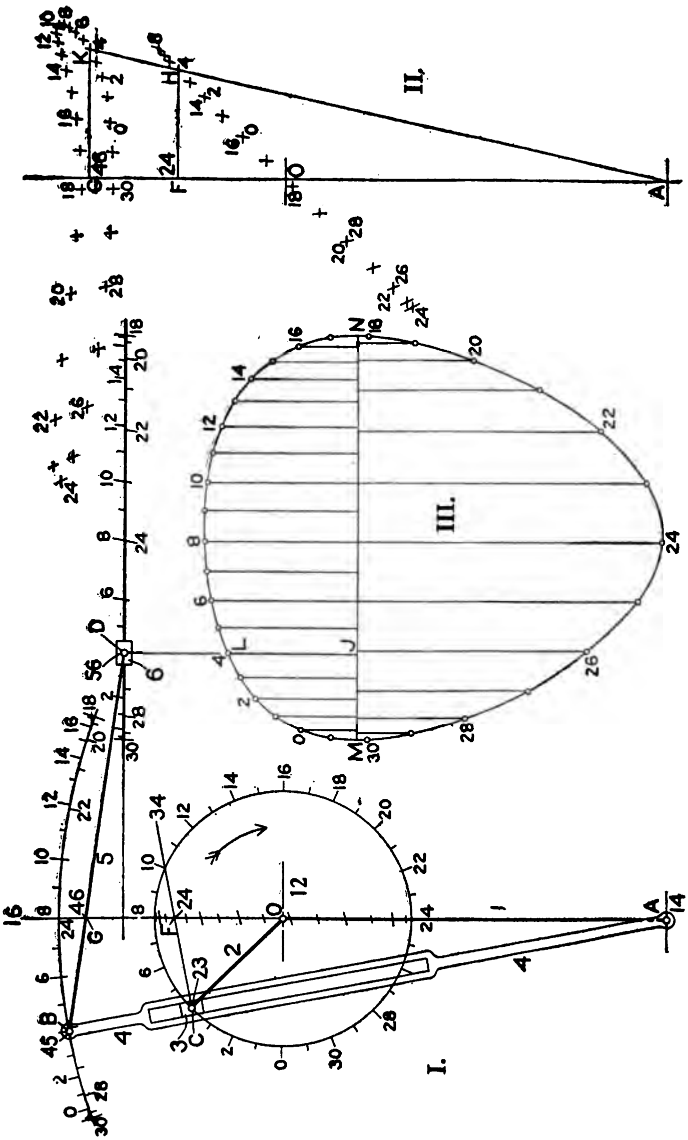

FIG. 40.—Analysis of Crank and Slotted Lever Mechanism.

velocity of the slide 6. In diagram III, positions of D are projected down from the stroke-line, and GK is laid off at JL.

Note the following points in regard to the construction of these diagrams:

The circle of the driving crank OC is divided into equal parts, here eight to the quadrant.

The motion of the lever AB is symmetrical, hence the construction for F need actually be made for one side only.

With OC and OF representing velocities to scale, the locus of H is a line at an angle of 45 deg. with AG. For any other scale, the angle would be different, but F would still be projected from I to II, and FH would now be determined as an intercept.

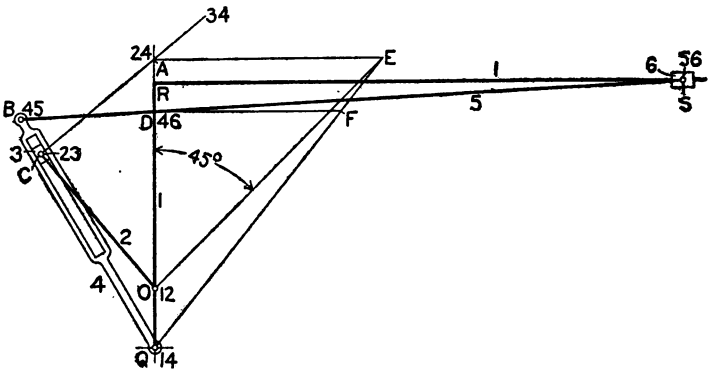

FIG. 41.—Whitworth Quick-return Motion.

The motion of the slide is not symmetrical, hence the height of G or 46 must be found for every position of the mechanism.

Diagram III is doubly useful, in that it shows both position and velocity, the successive positions being separated by equal intervals of time.

45. The Whitworth quick-return motion is outlined in Fig. 41: to the mechanism of Fig. 39*d* are added rod 5 and slide 6. The arm 2 rotates as a crank, at uniform speed: links 4, 5, 6, and 1 constitute an offset slider-crank with non-uniform velocity for crank 4. The determining points of common velocity are again

the centers 24 and 46, and they are derived just as in Fig. 40. For piece 2, if OC is velocity of C, OA will be velocity of A: for piece 4, AE equal to AO being the velocity of A, line QE will cut off DF as the velocity of point D and of slide 6. As at II in Fig. 40, we might change the velocity-scale, whereupon OE would make an angle other than 45 deg. with AO, but E would still be located by a horizontal line (a perpendicular to QOA) from 24 or A. In an actual drawing, the proportion diagram AE–QE–DF should be separated to avoid confused overlapping, as is done in Fig. 40.

46. The simple method outlined in Fig. 41 has the drawback that when arms 2 and 4 are near the horizontal, or slide 6 near

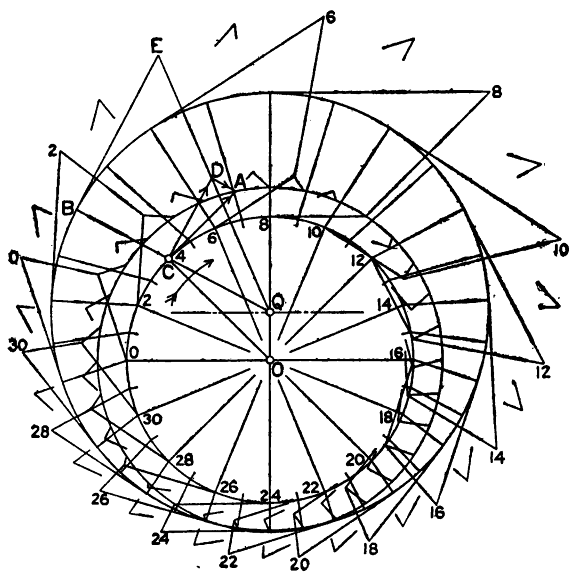

FIG. 42.—Velocity of Driving Pin.

the ends of its stroke, the intersection 24 or A will run off the paper, so that it becomes impossible to locate E and fix the inclination of QE exactly. Practically, this difficulty is not of much importance, for it affects only the small velocities near end

of stroke, point D or 46 being then near to Q; and the slant of QE can be so closely guessed at that the error in DF will be very small. It is, however, a matter of interest and of some importance to have an exact method that can be kept within the bounds of the drawing, and this is developed in Figs. 42, 43, and 44.

47. In Fig. 42, let CA (constant and perpendicular to OC) be the velocity of pin C. Resolve it into CD perpendicular and DA parallel to QC—note that the points marked by the several letters are the same as in Fig. 41, but that the centers O and Q are interchanged in relative position. Now CD will be the tangential velocity of point C on arm QB, while DA will be the velocity of block 3 in its slot (Fig. 41). By means of the usual proportion

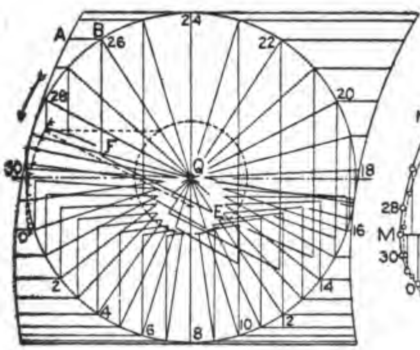

FIG. 43.—Determination of Velocity.

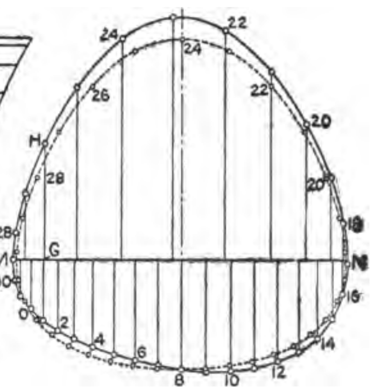

FIG. 44.—Velocity Diagram.

diagram, DE being a line from center Q, the velocity of point B is determined as BE. Since CA is constant, the point A will lie on a circle with center at O, and its successive positions can simply be stepped off from the C-points, without the trouble of drawing a perpendicular each time.

In Fig. 43, center Q is the same as in Fig. 42, and QB has the same set of positions. Each velocity BE is now measured radially inward from B. A regular piston-position diagram like Fig. 11 is drawn, for slide 6 in terms of crank 4 or QB, together with a rod-angle diagram similar to Fig. 36. Fig. 42 is drawn with

rotation in the clock-wise direction; but in Fig. 43 the direction is reversed, in order to get the arrangement usual on a shaping machine, which has the slow-cutting stroke in the direction away from the crank, at the same time keeping the slide to the right of the crank in the drawing: the offset of the stroke-line is one-half the crank-length QB, and the rod is three cranks long. Now EF, parallel to the rod, will cut off BF as the velocity of the slide, BF being perpendicular to the direction of stroke—compare Fig. 34. Finally, with slide-travel AB as abscissa at MG, and velocity BF as ordinate at GH, the velocity diagram in Fig. 44 is plotted and drawn.

The dotted curve in Fig. 44 shows the conditions that would exist if point B drove slide 6 through a cross-slot as in Fig. 2, or with "infinite" rod and central drive. The difference between the curves is due to the short rod and the offset of the stroke-line. The arrangement here discussed, with variable crank-velocity plus offset, may fairly be called the most general case of motion of the slider-crank mechanism.

DRAW PLATE 8.

J. Reducing Motions

48. In using the engine indicator, it is necessary to have some device for giving a reduced copy of the motion of the piston or crosshead of the engine. One point on this "reducing motion" or "indicator rig" is attached to the crosshead, to another is fastened the cord that moves the paper-drum of the indicator; and for a correct reduction there should be a constant ratio between the velocities of these two points. We shall now consider several linkage mechanisms used for this purpose.

Fig. 45 shows the pantograph, which is very simple in a kinematic sense, and gives a geometrically correct reduction. It is, essentially, a jointed parallelogram, with the sides extended as necessary: if a straight line is drawn across this linkage, locating three points on the center lines of three of the links, these points will remain on a straight line and the distances between them will

remain in a constant ratio. In Fig. 45 we see that the triangle ADC has the line EB kept always parallel to the base AD; consequently, the variable side AC will always be divided in the same ratio at B that the constant side DC is at E. Point C is a pin on the crosshead, while at B there is a pin for the cord; and the constant value of the ratio AB/AC meets the requirement for true reduction.

In Fig. 46 are given typical forms of the pendulum rig, which, as nearly as possible, uses the simple lever for a reducing device. At I we see the pin-and-slot drive, at II a link connection at the crosshead. The cord may be led off from a pin, as at I and II,

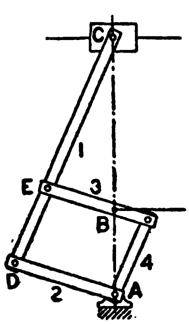

FIG. 45.—Pantograph.

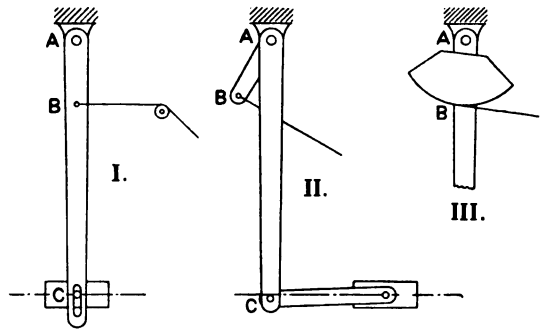

FIG. 46.—Pendulum Rigs.

care being taken that arm AB is perpendicular to the cord when AC is perpendicular to the stroke-line of the engine; or a sector may be used, as at III, to keep the arm AB constant in length. In general, this rig is only approximate in its reducing acting, and we shall now analyze it kinematically, in order to see how large are the errors and what is the range of effective usefulness.

49. A full discussion of the first device in Fig. 46 is illustrated by Fig. 47. At I the mechanism is outlined, with the pin and slot expanded to the full kinematic equivalent of a turning and a sliding pair. The instantaneous center between crosshead 2 and lever 4 is at 24, the intersection of the lines 12–14 and 23-34. As the pendulum swings away from mid-position, the point of common velocity at C gets farther from A, or the effective

length of the lever increases, and hence its rate of angular motion decreases.

The influences of these changes is shown by the curves at II, which are plotted on the stroke-line of the crosshead 2 as a base; the successive ordinates represent the center-line AC of I, moved out to equally-spaced positions of the point 23, on each side of mid-stroke. Curve CC is a locus of the instantaneous center C,

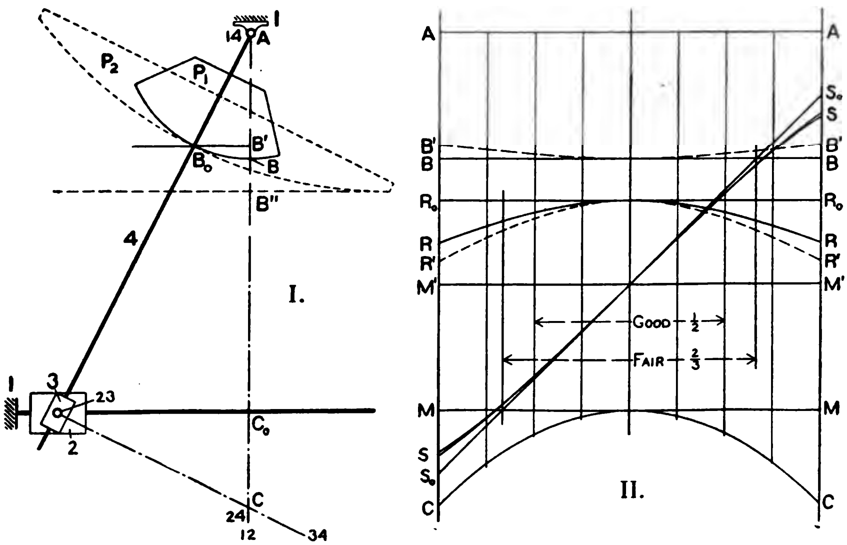

FIG. 47.—Analysis of Slot-and-pin Pendulum Rig.

and BB a similar locus for the cord point B (the point where the cord crosses the center-line). The straight line BB is for the case of " pulley " P_1, with radius from A; curve $B'B'$ shows how the effective lever-arm of the cord varies when it is fast to a pin at B_0; in either case, the cord is supposed to remain always horizontal.

The velocity-ratio AB/AC is now plotted from MM as base, to a convenient scale; MR_0 is equal to AB/AC_0, and the line R_0R_0 represents the ideal case of maintaining this ratio constant. Curve RR shows AB/AC, and curve $R'R'$, AB'/AC; it is evident

that an arc is better than a pin for the cord. The dotted outline P_2 is a "pulley" so laid out as to make AB″ bear a constant ratio to AC, and thus realize R_0R_0 as a ratio "curve." The vertical movement of the point B″ will be so considerable with this arrangement that it will appreciably change the inclination of the cord, thus complicating the problem of getting a correct profile for arc P_2.

Finally, the actual displacement of the cord, or the movement of the surface of the indicator drum, is represented by the curves SS. The ordinate is either the distance B_0B' or the arc-length B_0B (to an enlarged scale), while the base-line M′M′ represents mid-position of the whole mechanism, with B_0 on the line AC_0: the two cases of arc and pin cord-drive are distinguishable only near the ends of the stroke. The straight reference-line S_0S_0 shows what the movement would be if the ratio R_0 were maintained throughout the stroke, as by the use of arc P_2. With a piston-stroke equal to about half of AC_0, the curve SS is practically a straight line, with its mean ratio a little less than R_0. The stroke should never be more than two-thirds of the pendulum length AC_0, or the pendulum less than one and one-half times the stroke.

50. Similar diagrams for the pendulum rig with link connection are given in Fig. 48, where the abscissa is displacement of the driving-pin 23 (on the crosshead) from mid-position. The effective lever-length AC is determined by the intersection of the center-line of link 3 with the vertical center-line AC. As shown by the R-curves, this mechanism gives a nearly constant ratio of reduction over a good range of movement, but drops away rapidly toward the extremes, especially to the right. In terms of the nominal pendulum-length AM, and with the proportion used in the figure, the cord-pin gives the truer reduction, over the range from $\frac{1}{4}$ right to $\frac{1}{4}$ left; the arc is not quite so correct, but covers effectively the wider range from $\frac{1}{4}$ right to $\frac{1}{2}$ left. A special exact arc, similar to P_2 in Fig. 47, can be laid out for this mechanism also; but if we try to carry the movement too far toward the right, we come to a point beyond which, in order to keep the cord-arm AB in true ratio to the rapidly shortening AC, the cord-direction

on the plane of the pendulum will cut across a curve tangent to those preceding it; so that true reduction becomes impossible. If a very short pendulum must be used, the range cannot be kept

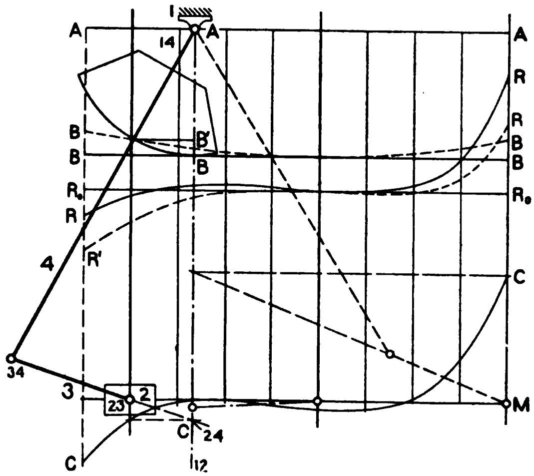

FIG. 48. The Link-driven Pendulum.

symmetrical with reference to the vertical position of the pendulum, but must be carried farther toward what is the left side in Fig. 48.

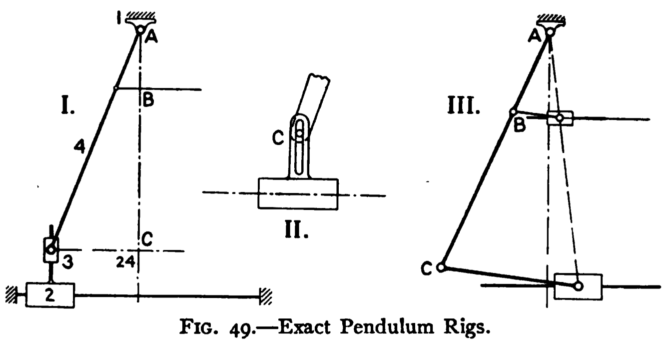

FIG. 49.—Exact Pendulum Rigs.

51. The mechanism at I in Fig. 49 is geometrically correct, provided that the cord remains horizontal—or practically parallel to a mean position located as at II in Fig. 46. The equivalent

pin-and-slot drive is shown at II; it is simpler to make and good enough for occasional use, but for durability the guide-bar and slide-block shown at I are decidedly preferable. Any arrangement such as is outlined at III, where the mechanism that takes from the pendulum the motion for the cord is geometrically similar to that which gives to the pendulum its motion from the crosshead, is, of course, correct.

DRAW PLATE 9.

K. Gear-Tooth Profiles

52. The diagrams shown in Figs. 50 and 51 are intended partly to give practice in drawing the profile curves, partly to enforce certain principles in regard to the action of gear teeth. In Fig. 50, the cycloidal curves are drawn by holding the pitch circles A and B fixed, and rolling the describing circles C and D upon them, thus getting the set of curves originating at the pitch-point P. For P1, circle C rolls on pitch-arc PG, for P2 it rolls on PH; for P3, circle D rolls on PK, for P4 it rolls on PL. These tooth curves are extended to a distance (radially, from the pitch circle) about equal to the radius of the rolling circle—this being a good deal more of the curves than is used in actual wheels. Near the beginning of approach, a pair of profiles are drawn with contact at E, near the end or recess a similar pair with contact at F.

In Fig. 51 the same general procedure is followed, the involute curves which touch at P being first constructed, then moved out to contact at E and at F. Below the base circles, the flank profiles of the teeth are radial straight lines.

53. The best way of reasoning out the solution of the problem of generating proper tooth profiles, illustrated in Figs. 50 and 51, is as follows:

Consider Fig. 50 first. Let pitch circles A and B and describing circles C and D all turn about fixed centers on the line AB, rolling on each other at the point P. On the circumference of C take a certain describing point E; as circles A and C roll together, point E will trace a curve ES on the moving plane of wheel A;

and as B and C roll together, point E will trace the curve ET on the plane of wheel B. Since these two cycloidal curves are being traced by the same moving point at the same time, they will have a common point at each successive instantaneous position of E, with perfect tangency at that point. The action of point F in tracing profiles QF and RF is exactly similar.

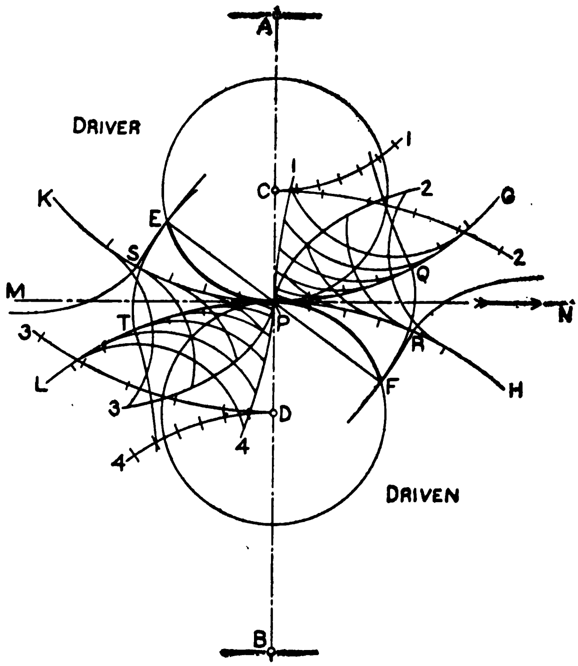

FIG. 50.—Cycloidal Tooth Profiles.

It is evident that the arc EP is a locus, fixed in position, of the contact-point E, and PF a locus of F. Further, P is the common instantaneous center of A, B, and C, or of A, B, and D; consequently, the radius EP is, at any instant and at the same time, rotating about the point P on the plane of wheel A and on the plane of wheel B (with different relative angular velocities); hence this radius is a common normal to the two profiles, and is

therefore the line of thrust, or of pressure of tooth upon tooth. Note that with cycloidal curves this line is continually changing directions, swinging back and forth over a range form the tangent MN to a limiting position of EP which is determined by the addendum of the teeth

54. With involute teeth, Fig. 51, the pitch circles have no direct part in the operation of generating profiles; the elements involved are the two base circles and the tangent-line CD, which

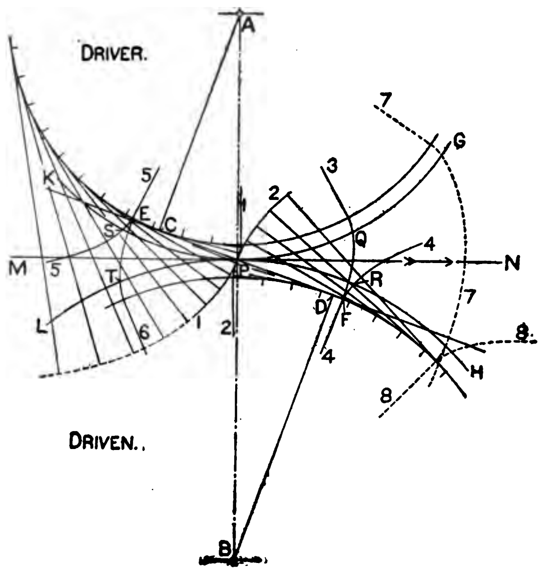

FIG. 51.—Involute Tooth Profiles.

rolls on these circles without slip. A describing point on this line simultaneously generates two curves, on the respective moving planes of the two wheels, and these curves satisfy the requirement of smooth, continuous contact as the pitch circles roll on each other. Line CD is the locus of contact, and points C and D are the respective instantaneous centers of base circle AB with line CD and of base circle BD with line CD.

The dotted extension of curve P1, with the dotted curves at

the right of Fig. 51, are intended to illustrate a discussion of the profile below the base circle—see Problem 6.

55. It is self-suggestive that other describing curves might be used, beside the circle and the straight line, for generating tooth profiles; and it is entirely true that these are only the simplest cases out of a wide range of possibilities. They are, however, the only ones used in ordinary practice, and there is no need to take up a general discussion of the subject here.

DRAW PLATE 10.

APPENDIX

Directions and Data for Drawings

A sheet 19″ by 24″ will be large enough for these plates. Rule a one-half inch margin, leaving a working surface 18″ by 23″.

Drawings are to be neatly finished in pencil, but not inked. Use a simple freehand letter for titles and notes. Draw accurately, using a hard pencil kept well sharpened.

PLATE 1.—PISTON DISPLACEMENTS.

FIG. P 1.*—Crossed Slide.
FIG. P 2.—Engine Mechanism.
FIG. P 3.—Developed Diagrams.

Figs. P 1 and P 2 are like Figs. 3 and 6, Fig. P 3 like Fig. 7. Divide circle at every 10 deg., numbering divisions at every 30 deg. Lower part of Fig. 6 shows finished form of Figs. P 1 and P 2; on Fig. P 3, draw ordinates as on Fig. 7. For Fig. P 2, show only the portion of Fig. 6 within the limits $A_1B_1B_2A_2$. Draw harmonic motion curve on Fig. P3 in dotted line.

Take crank-radius $R=3.5$ in., rod-length $L=10.5$ in., or ratio 3 to 1. In Fig. P 3 use for the base-line the scale 1 in. = 20 deg.

To divide a circle in 10-degree parts, first locate the quadrant points accurately, then strike off the radius on each side of both of these points, dividing the circumference into 30-degree parts. Check these for realized equality, using the large dividers, then

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* Letter P is here prefixed to plate figures to distinguish them from those in text. Omit this letter on the drawings.

trisect one by trial with the spring-bow dividers, and carry this sub-division all around the circle.

PLATE 2.—PISTON AND CRANK POSITIONS.

FIG. P 4.—Crank Angle for Piston at each One-tenth of Stroke.

FIG. P 5.—Polar Diagram.

For Fig. P 4, combine Figs. P 1 and P 2, or Fig. 3 and 6, in one diagram, but without angle divisions; use dotted lines for first case, full lines for second. Divide the stroke-line into 10 equal parts, and find the crank position corresponding to piston at each one-tenth of the stroke, also to $^1/_{20}$ and $^{19}/_{20}$. Use upper

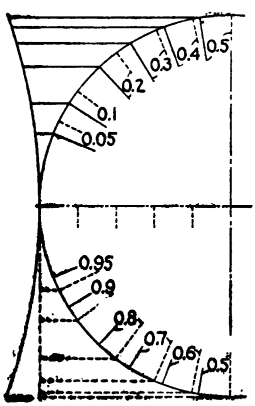


FIG. 52.—Notation for Fig. P 4.

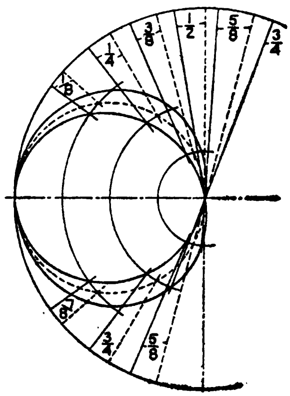


FIG. 53.—Notation for Fig. P 5.

half of figure to make determinations for actual mechanism, lower half for the case of infinite connecting-rod, but carry each set of divisions all the way round the circle, with the scheme of marking shown on Fig. 52.

For Fig. P 5, draw same primary diagrams as in Fig. P 4, put in radii at each 10 deg., and plot polar curve like Fig. 7. Do not number degrees, and let all the construction for the polar curve appear in faint lines. Draw the harmonic-motion curve as a circle, merely trying the plot of a few points to confirm the

geometrical proof of Fig. 9. On this diagram, find the crank positions for the piston at each one-eighth of the stroke. Indicate these positions by heavy radial lines, dotted and full for the respective cases, as in Fig. 53. Note how this diagram lacks accuracy in the determination of crank position near mid-stroke.

Use $R=4$ in. and $L/R=3$. This size of crank-circle will be used on several of the following plates, and it will be well to make a template of the division, to save a fresh laying out each time.

On Fig. P 4, locate the crank-pin positions by measuring from the limiting arc A_1A_2 or B_1B_2, setting the scale against the T-square to keep it horizontal, and moving it up and down till the desired distance is secured.

PLATE 3.—The Offset Stroke Line.

Fig. P 6.—Direct Diagrams.
Fig. P 7.—Equivalent Straight Mechanism.

Use same dimensions as on Plate 2, with stroke-line offset 4 in. upward.

Fig. P 6. Lay out mechanism as in Figs. 10 and 11, including stroke-line MN and the lines OM, ON, and OP, the last as on Fig. 12, to fix the mid-point P and the mean rod-line KL of Fig. 13. Draw the tangent arcs of Fig. 11 (full-line) and the straight limit-lines of Fig. 13 (dotted). Divide the circle at every 10 deg., using K as the zero-point, not A. Draw horizontals out to the limit-lines, as on Fig. 6, also inward to a mid-stroke line got by drawing through the center O a line perpendicular to KL. Number at every 30 deg.

Fig. P 7. By means of triangle KOA′ of Fig. 14, find radius OA′ of the crank for an equivalent plain crossed slide. With this new radius R' draw a harmonic movement diagram (Fig. P 1), also the diagram for an ordinary engine mechanism (Fig. P 2), keeping the rod ratio 3. Divide for each 10 deg., with A as the zero-point. Note that only half of the figure need be drawn, above the center-line AB, as the movement is symmetrical with respect to this line.

PLATE 4.—PISTON MOVEMENTS COMPARED.

FIG. P 8.—Comparison on Developed Circle.

Draw a diagram like Fig. 7 in general scheme, with the same scale of base, 1 in. = 20 deg. Put the lines AA, BB at the distance R (the actual crank) from the center-line; then all curves will run outside of these lines. Plot curves from Figs. P 6 and P 7 as follows:

a. Harmonic motion curve. Take ordinates from both Fig. P 6 (method of Fig. 13) and Fig. P 7 (equivalent crossed slide), and see that they are identical. Dotted curve.

b. Actual motion. Take ordinates from Fig. P 6, measuring them from the mid-stroke line outward, and plot them from the center-line OO. Full-line curve.

c. Plain slider-crank, of same nominal stroke—considering the length A′B′ on Fig. 13 as the "nominal" stroke for Fig. 10, not quite the same as MN. Take ordinates from Fig. P 7. Dot-and-dash curve.

In drawing these curves on the same base-line, we eliminate the angle KOA′ or γ of Fig. 13; that is, the zero of angle, at OK on Fig. P 6 and at OA′ on Fig. P 7, is made the same on Fig. P 8.

Curves *b* and *c*, thus plotted on the same base, give the best comparison that can be made between the two mechanisms: the differences between the two curves show differences in the effect of rod-swing in the two cases. As the rod is longer, relative to the crank, these differences will be less.

PLATE 5.—MOVEMENT OF CORLISS VALVE.

FIG. P 9.—Layout of Mechanism.

FIG. P 10.—Curves of Valve Movement.

a, Corliss Gear;
b, Harmonic Motion.

FIG. P 9. Construct Diagram like Fig. 16, using dimensions on Fig. 15 (unless others are assigned). Omit circle on AB and arc EE. Draw figure half-size, with KV full-size.

FIG. P 10. From Fig. P 9, plot diagram like Fig. 17, using for the base the scale 1 in. = 30 deg.

Problem 2 or 3 may be substituted, if desired or directed.

PLATE 6.—VALVE DIAGRAMS.

FIG. P 6.—Valve and Piston Diagrams.

FIG. P 12.—The Valve Ellipse.

Draw diagrams like Figs. 26 and 27, using either Reuleaux or Zeuner valve diagrams as preferred or directed. Lay out only the full-line curve of Fig. 27. Put in all the lap-lines—for both ports.

For the valve gear, use $r=2\frac{1}{2}$ in., angle $\delta=135$ deg., steam-lap $s=1\frac{5}{8}$ in., exhaust lap $i=\frac{1}{2}$ in. For the engine mechanism, let 4 in. represent R, and make $L=5R$. Divide at every 10 deg. and number at every 30 deg., as on preceding plates.

PLATE 7.—PISTON VELOCITY.

FIG. P 13.	Central Stroke-line.	FIG. P 14.
For Crank Angle.		For Piston Position.
FIG. P 15.	Offset Stroke-line.	FIG. P 16.

Figs. P 13 and 14, like Figs. 37 and 38, with same omission of lower part of circle.

Figs. P 15 and 16, similar diagrams for the offset stroke-line, with full circle.

Take crank 4 in. long, rod 12 in. long, offset of stroke-line 5 in.

PLATE 8.—CRANK AND SLOTTED LEVER MECHANISM.

FIG. P 17.—Layout of Mechanism.	(I)
FIG. P 18.—Proportion Diagram.	(III)
FIG. P 19.—Velocity Diagram.	(III)

Follow Fig. 40, using the following dimensions:

AO = 8 in.;	AB = 12 in.;	AE = 11 in.;
OC = 2.5 in.;	BD = 8 in.	

Let OC represent the velocity v of crank-pin C. Use same notation and marking as on Fig. 40. Make eight divisions to the quadrant. Mark set of positions for point G.

As alternate problems, draw diagrams for the Whitworth motion, either by the method of Fig. 41 (arranging work as on Fig. 40), or by the method of Figs. 42 to 44.

PLATE 9.—REDUCING MOTION.

FIG. P 20.—(*Title to suit*).

Take one of the following problems, as directed:

a. Diagram like Fig. 47; $AC_0 = 12$ in., $AB = 4$ in.

b. Diagram like Fig. 48; link $4 = 12.5$ in., $AM = 12$ in., link $3 = 4$ in., $AB = 4$ in.

c. Design arc P_2 for Fig. 47, with dimensions as in Problem *a*.

d. Design exact arc for Fig. 48, with proportions as in *b*.

In the last two problems, have the cord run over guide pulley located near edge of paper.

PLATE 10.—GEAR TOOTH PROFILES.

FIG. P 21.—Cycloidal Profiles.

FIG. P 22.—Involute Profiles.

Draw diagrams like Figs. 50 and 51. Take $PA = 6$ in. and $PB = 8$ in.; $PC = PD = 2$ in.; angle of involute generating line, 20 deg. from MN. Use half-inch divisions; put Q, R, S, and T (Fig. 50) and E and F (Fig 51) at 5 divisions or 2.5 in. from P.

Problems

1. In the oscillating engine, the same slider-crank mechanism is used as in the regular engine, but the piece which is ordinarily the connecting-rod is now held fast as the frame. Prove that the piston-position diagram outlined in Fig. 54 is correct, formulating the method of drawing this diagram. Compare with Fig. 6; see also Fig. 39.

2. Steam-valve drive of Fleming four-valve engine, Fig. 55.

3. Steam-valve drive of Ball four-valve engine, Fig. 56.

In both these gears, the point A is driven from the eccentric, with harmonic motion in the horizontal direction. Fig. 55 has the same elements as the Corliss gear, bent lever 2 acting as a local wrist-plate. In Fig. 56, pieces 2, 3, and 4 constitute the moving parts of a four-link mechanism, and motion for the valve

is taken from the middle point of link 3, through the short rod 5. In working out motion diagrams for these gears, proceed as follows:

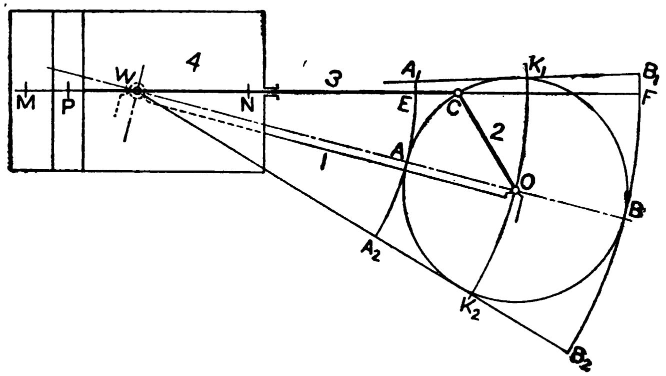
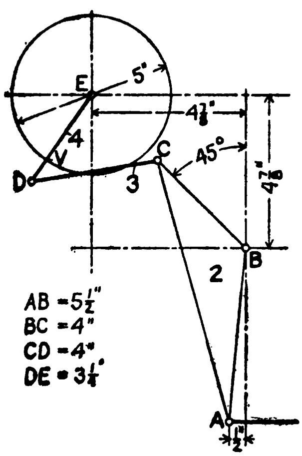


FIG. 54.—Piston Position in Oscillating Engine.

First, start with the point A, locating it at a number of positions spaced equally in the horizontal direction, and lay out the mechan-



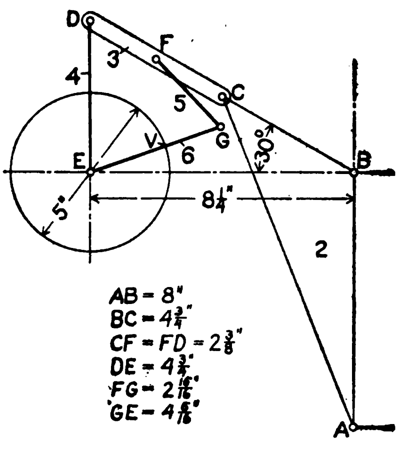


FIG. 55.—Fleming Steam-valve Drive. FIG. 56.—Ball Steam-valve Drive.

ism for each of these positions, after the manner of Fig. 16. With the movement of A as a base, plot a curve, showing travel of the

point V on the valve profile, to each side of the mid-position at which the mechanism is here drawn. Now choose some effective eccentric-radius for the point A—in both these engines this radius is variable, under the control of a shaft governor: for equally spaced positions of the eccentric-center, locate A on its base-line, and get the corresponding ordinate for travel of V. Finally, plot a curve like Fig. 17. For Fig. 55, let A have a full range of 6 in., then take an eccentric of 2.5 in. radius; for Fig. 56 let the corresponding dimensions be 10 in. and 4 in.

4. Design a pantograph like Fig. 45, to be used with an engine of 48 in. stroke, and to give an indicator diagram 4 in. long. Center-line of engine is 32 in. above floor, and reducing motion is to be supported from floor; include a suitable support, which can be made of wood, as can also the pantograph itself.

5. Design a slide-movement indicator rig of one of the forms shown in Fig. 57. In either case, P is a fixed pivot and C is a pin on the crosshead; the curved bar AB remains in contact with

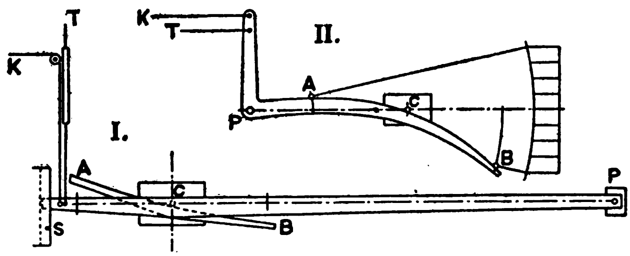


FIG. 57.—Slide-movement Indicator Rigs.

pin C as the crosshead travels back and forth, and the cord K is thus given a motion in constant ratio to that of the crosshead. For both arrangements, let the stroke be 12 in.; take PC as 12 in. in I and as 30 in. in II; design for a cord-travel of 3 in.

Note that in these problems, as in problems *c* and *d* under Plate 9, the method to be followed is closely analogous to that used in getting the point P″ on the centrode for the connecting-rod in Fig. 30.

6. Following the indications of the dotted curves on Fig. 51, investigate what happens if, with involute profiles, contact is carried beyond the points C and D, where the describing line (and locus of contact) is tangent to the base circles. As the describing point passes the position D, it begins to trace on wheel B a reversed involute; and the extended involute on wheel A, which has perfect contact with this new curve, cuts across the old profile FR. Verify these statements by actually making the drawing, and derive conclusions as to the limits of usability of the involute profile.

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